COMBUSTION

NOV 27 1935

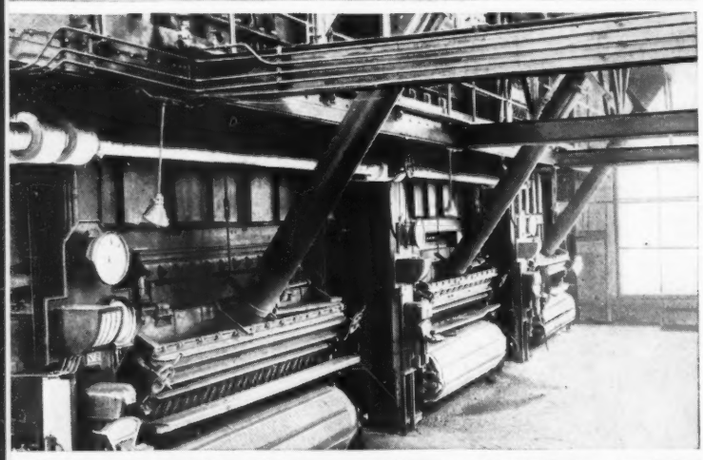
DEVOTED TO THE ADVANCEMENT OF STEAM PLANT DESIGN AND OPERATION

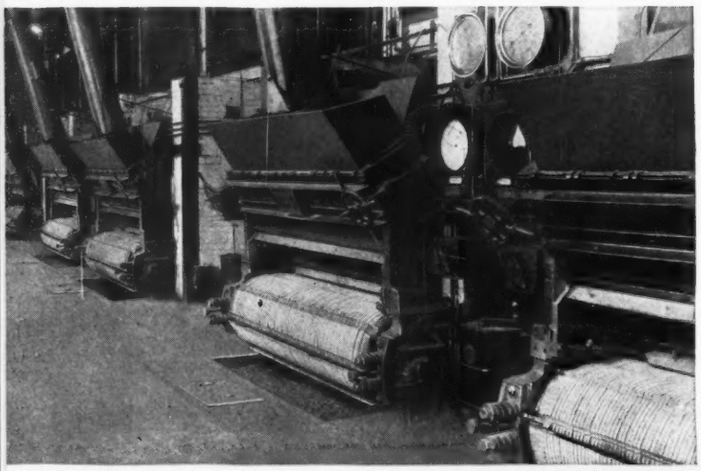
Vol. 7, No. 5 NOVEMBER, 1935 25c a copy

Boiler Room of Chevrolet Motor Company, Baltimore, Md.

Modernizing the Conners Creek Power Plant

Boiler Design as Influenced by Factors External to Heat Absorbing Surface





10,500,000

sq ft of boiler heating surface has been equipped with

COXE

TRAVELING GRATE STOKERS

and

52 PER CENT

of all Coxe Stokers sold to date have been purchased on

REPEAT ORDERS

For optimum results with small sizes of anthracite, coke breeze or lignite, specify
COXE STOKERS

(Top) Coxe Stokers installed to burn No. 3 buckwheat under 4540 sq ft sectional header boilers in the plant of the American Museum of Natural History, New York.

(Upper middle) Coxe Stokers installed to burn coke breeze under 6000 sq ft sectional header boilers in the Hunts Point plant of the Consolidated Gas Company, New York.

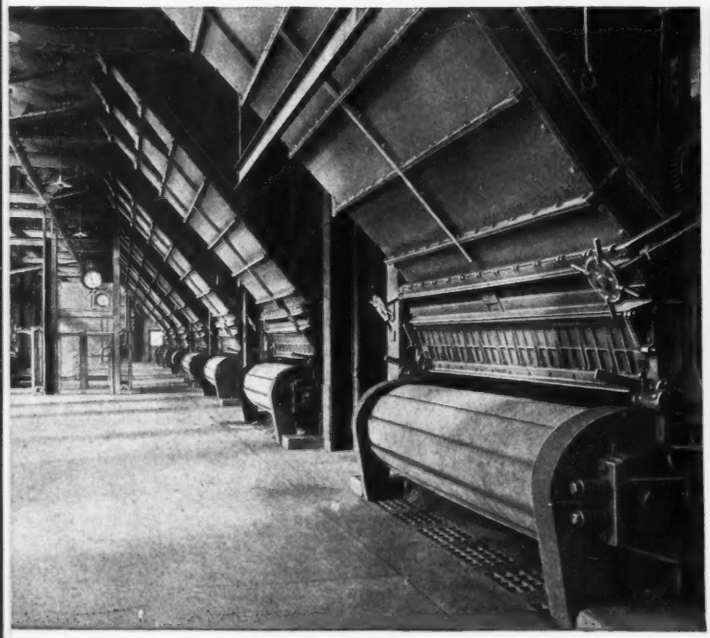
(Lower middle) Coxe Stokers installed to burn No. 3 buckwheat under 3000 sq ft sectional header boilers in the plant of the Remington Arms Co., Inc., Ilion, N. Y.

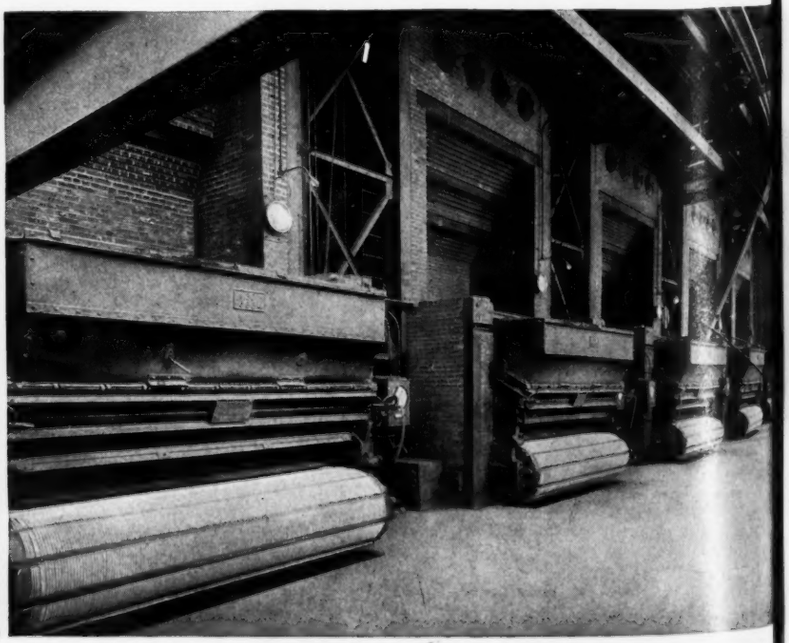
(Lower Left) Coxe Stokers installed to burn No. 4 buckwheat under 6680 sq ft bent tube boilers in the Nanticoke, Pa., plant of the Glen Alden Coal Company. This company has installed in thirteen of its plants a total of 105 Coxe Stokers furnished on 34 orders.

(Lower right) Coxe Stokers installed to burn No. 3 buckwheat under 6120 sq ft bent tube boilers in the plant of Cornell University, Ithaca, N. Y. Three orders, covering an aggregate of seven stokers have been received from this plant.

COMBUSTION ENGINEERING COMPANY, INC.
200 Madison Avenue, New York, N. Y.

Canadian Associates: Combustion Engineering Corporation, Ltd., Montreal





COMBUSTION ENGINEERING

COMBUSTION

DEVOTED TO THE ADVANCEMENT OF STEAM PLANT DESIGN AND OPERATION

VOLUME SEVEN NUMBER FIVE

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FOR NOVEMBER 1935

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H. STUART ACHESON, *General Representative*

ALFRED D. BLAKE, *Editor*

THOMAS E. HANLEY, *Circulation Manager*

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CCA

What are the pertinent facts controlling boiler feedwater treating costs?

The influences governing low treating costs are—

First—The amount and quality of the contained water impurities.

Second—Kind of treatment employed.

Third—Scope or extent of the conditioning practices employed.

Fourth—Exactness with which all treatments are made to meet changing water and operating conditions.

The First classification needs no special discussion, because it is logical to assume that as impurities in a water increase or decrease the corrective agents required for their removal must increase or decrease in the same proportion. For example, if a water contains one grain of calcium sulphate (terms $CaCO_3$), approximately one grain of soda ash or $2^1/_2$ grains of trisodium phosphate removes it. Similarly, two, three, four, etc. grains of calcium sulphate or its equivalent in a raw water require two, three, four, etc. times these same treatments.

The Second classification is probably best understood by reciting some actual experiences.

Case I—an example where the incorrect type of softener was selected. Here, zeolite equipment was installed *alone* to condition a water containing,

13.1 grains calcium bicarbonate, (terms $CaCO_3$)
1.3 grains magnesium bicarbonate, (terms $CaCO_3$)
2.03 grains magnesium sulphate
0.14 grain magnesium chloride
0.41 grain silica

The zeolite treating cost in this case approximated 10 cents per 1000 gallons, including treatments for sulphate ratio adjustments and for removing the final hardness with phosphates. A lime-soda hot-process softener costs, for the same water, would have approximated 2.25 cents per 1000 gallons with similar phosphate additions. The high bicarbonate of soda in the zeolited water, in causing the delivery of large volumes of carbonic acid to the steam, brought about extensive repairs in this case to condensate return lines, all chargeable to treating costs. *Such misjudgments in the selection of a water softener set-up are absolutely inexcusable when the judgment of experienced and unbiased water purification engineers is always available for consultation purposes.*

Case II—an example where the installation of a softener was unnecessary. Here, a lime-soda hot-process softener operation was replaced by internal treatment. Before this change was made the chemical costs alone averaged 1.6 cents per 1000 gallons of finished water. After going to the internal treatment, this was cut to 0.46 cent per 1000 gallons with a resulting reduction in chemical costs of nearly 70 per cent and with excellent results. This is just another example of useless expenditures which could have been prevented through consultations with experienced water purification engineers.

Case III—where soda ash replaced a soda phosphate treatment. In this case, the silica in the deposits from the fire-exposed boiler tubes was near 5 per cent which was rather low for a phosphate treatment. A change was made to soda ash, which proved satisfactory and treating costs were reduced 67 per cent. *Feedwaters, under such circumstances, invariably contain calcium bicarbonate and the phosphates in use react in the formation of sodium bicarbonate which later goes to sodium carbonate or soda ash. This proves rather a costly practice when soda ash can be purchased in the open market at one-fourth the cost.*

Case IV—waste of money resulting from the use of materials of unknown composition and value. The following analysis on which this is based represents an average of fifty of these materials.

Caustic soda (NaOH)	2.7%
Soda ash (Na_2CO_3)	12.0%
Organic matter	5.8%
Phosphates (Na_3PO_4)	2.0%
Undetermined	1.4%
Water	76.1%

Based on the existing market price of these chemicals, the cost of 100 pounds approximates 78 cents.

As the user of such materials pays something like $12.00 for this same amount, the profit, sales expense and whatever service is extended takes $11.22 or $93^1/_2$ per cent of the sale price, leaving $6^1/_2$ per cent for preventing boiler scale, corrosion and the many kindred troubles resulting through the use of boiler feedwaters.

The use of such highly adulterated materials at any time cannot produce other than excessive treating costs when the active materials they contain can be purchased singly as commercial articles or in a compounded form without this large addition of water.

The Third classification or scope of a treatment on cost influences is understood best by assuming that soda ash is used singly to treat a water that causes—

Boiler scale and corrosion
Feed pipe and pump deposits and corrosion
Dirty steam

and it costs 2 cents for soda ash per 1000 gallons to treat in this way and $5^1/_2$ cents where all repair expenses caused by water impurities are included in this cost. Now, assume again that the use of chemicals for correcting all unsatisfactory conditions costs $3^1/_2$ cents and is increased to 4 cents with the repairs. The net saving in using a complete treatment is $1^1/_2$ cents per 1000 gallons of boiler feedwater.

It is invariably difficult to evaluate the actual savings through complete treatments because of the far-reaching effect of water impurities on repairs, but the example given is no exaggeration of the possibilities and actually underestimates the savings in cases where the qualities of the water in use are of a highly complex nature.

The Fourth classification is probably the *most important influence on treating costs, because it involves a testing set-up and experienced supervision* that guarantees the best treatment and keeps it on the job of

(a) Guarding the boilers, turbines, feedwater and steam lines, superheaters, pumps, etc., against deposits and corrosion to the maximum extent and, also, keeps chemical wastes to the possible minimum.

(b) Indicates these effects under actual operating conditions.

•

In summarizing this whole situation, the only logical way to maintain the best all-around results at the minimum cost in chemicals and repairs to equipments affected by water impurities is to use materials of known strength and effect and to the limits required to correct the conditions for which they are employed; to set up the controls for checking the results of such treatments under operating conditions; and to get the best advice on this.

An organization with no affiliations for profit in the sale of water treating equipments and materials and which possesses the experience and facilities for research work is again best equipped for extending advice in all such matters.

Cyrus Wm. Rice & Company, Inc., of Pittsburgh, with its twenty years of constant service to power plants, thirty-five years treating experience and excellent chemical laboratory facilities for all kinds of check and routine analyses is such an organization. During this period, it has been continuously employed by many of the most representative corporations in the United States and has handled treatment problems for boilers operating up to 1400 pounds pressure and 1000 per cent rating.

It is the "Daddy" of the present universal practices in use for testing boiler concentrates in the close treatment regulations for the prevention of boiler scale and corrosion and *for preventing wastes through excessive boiler blow-downs.*

It also started the practices of checking the steam quality and of regulating this quality through tests of the blow-down water. The test set-ups at the plants for these controls are very definite and within the understanding of any plant operator. This was all started thirty years ago.

In 1920, the same organization began the practice of registering the corrosive properties of feedwaters through the location within the feedwater loop of *sensitized pieces of boiler steel.* This was followed in a few years by the location of small plates of polished similar metals under the water line within the steam drum.

In the intervening fifteen years, these detectors have truthfully *registered the accumulative effect of any contained corrosive gases in the feedwater in a manner that leaves no doubt as to their value in this direction.* Besides this, the retention of the older rods provides a very definite record which shows the progress of a treatment in eliminating the corrosive gases. Not only this, *but this new way replaced the costly and unsatisfactory practice of periodically analyzing the feedwaters for oxygen and carbonic acid.*

For many years, this same organization *recognized the necessity* of developing some inexpensive treatment for *preventing the hard silica scale depositions* which phosphates in almost any concentrations failed to do. Laboratory work was started on this and over a year ago resulted in the production of *PHOSCALOID.* Since then it *has proven its exceptional value in preventing silica depositions at 80 per cent of the cost of commercial soda phosphates.* This practice of *cutting expenses* without detracting from the result has been a *policy of the Rice organization* since its inception because at no time during its many years of existence has it resorted to the use of other than commercial chemicals for water treating purposes, nor has it employed any but the best standard laboratory methods for controlling their use.

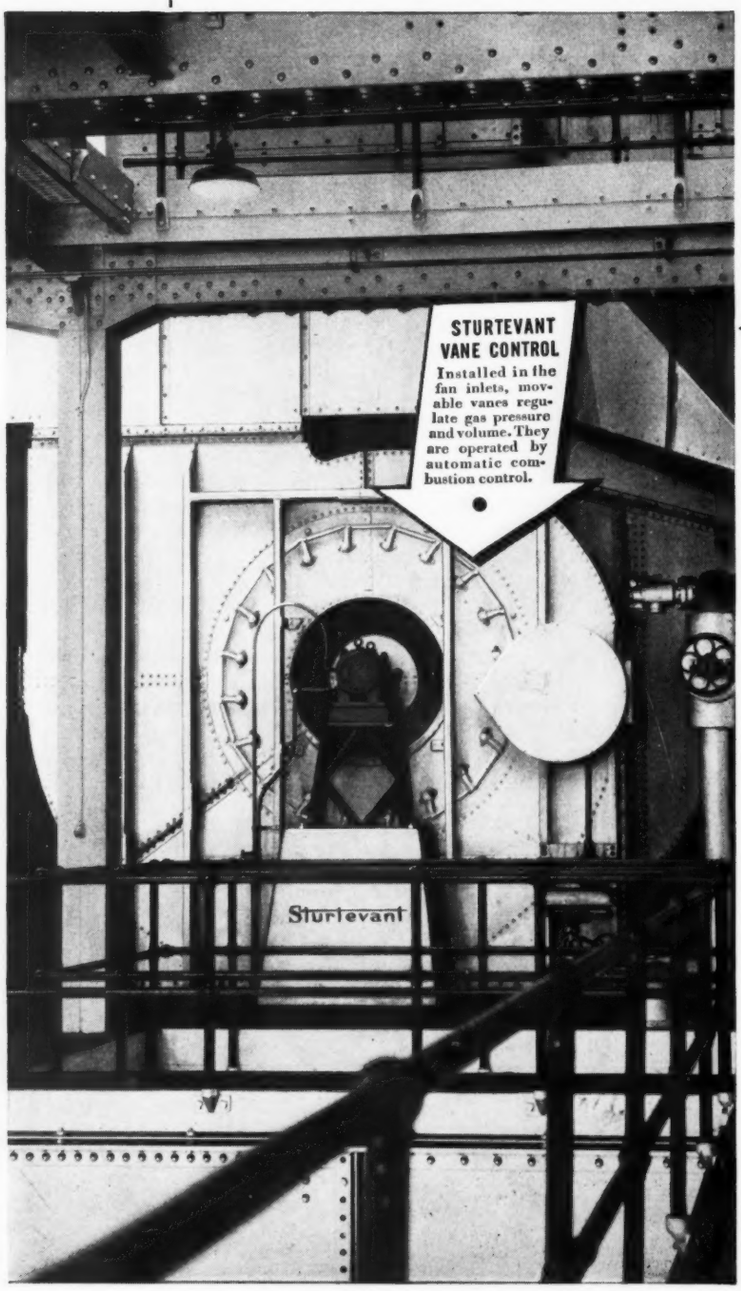
If high water treating costs and unsatisfactory results caused by water impurities are issues requiring attention at any one plant, *the best advice* for correcting them *is none too good,* because it does assure the most permanent conclusion to the solution of such problems by injecting reasoning into the work through the use of chemicals and tests of known value. It is the course which every wide-awake engineer wishes to see inaugurated.

Cyrus Wm. Rice

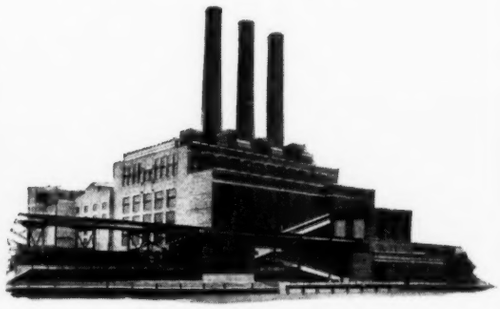
President

CYRUS Wm. RICE & CO., Inc.
HIGHLAND BUILDING—PITTSBURGH, PA.

PROFIT BY THIS EXPERIENCE WHEN SELECTING MECHANICAL DRAFT FANS


STURTEVANT VANE CONTROL
Installed in the fan inlets, movable vanes regulate gas pressure and volume. They are operated by automatic combustion control.
Sturtevant

One of 6 Sturtevant Vane Control Induced Draft Fans in the Ashtabula Station of The Cleveland Electric Illuminating Co.



Ashtabula Station
Cleveland Electric Illuminating Co.

When the great Ashtabula Plant of The Cleveland Electric Illuminating Company was designed...proved performance was the guide. Equipment which had made good in this company's highly efficient Avon Plant was specified...so exceptionally satisfactory had been the operation of this station.

Two years experience at Avon with Sturtevant Vane Control Induced Draft Fans led to the use of six of these same fans at Ashtabula. They serve boilers for three 50,000 K.W. units.

In selecting draft fans for your plant, why not profit from the experience of this utility and other of the country's most exacting purchasers of mechanical draft equipment?

B. F. STURTEVANT CO., Hyde Park, Boston, Mass.
New York, N.Y., 420 Lexington Ave. Chicago, Ill., 400 N. Michigan Ave.
San Francisco, Cal., 681 Market St. *Branch Offices in Other Cities*

B. F. STURTEVANT COMPANY of CANADA, Ltd., GALT.
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Sturtevant
REG U.S. PAT. OFF.
Draft Fans

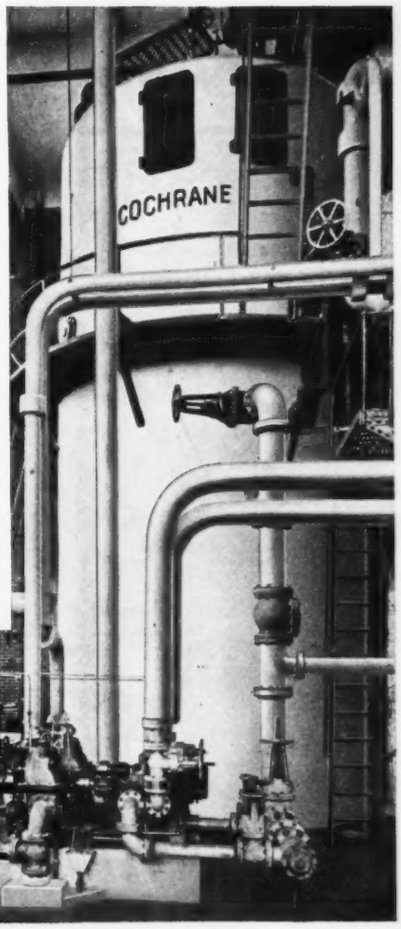
DRAFT FANS · TURBINES · GEARS · ECONOMIZERS · AIR HEATERS

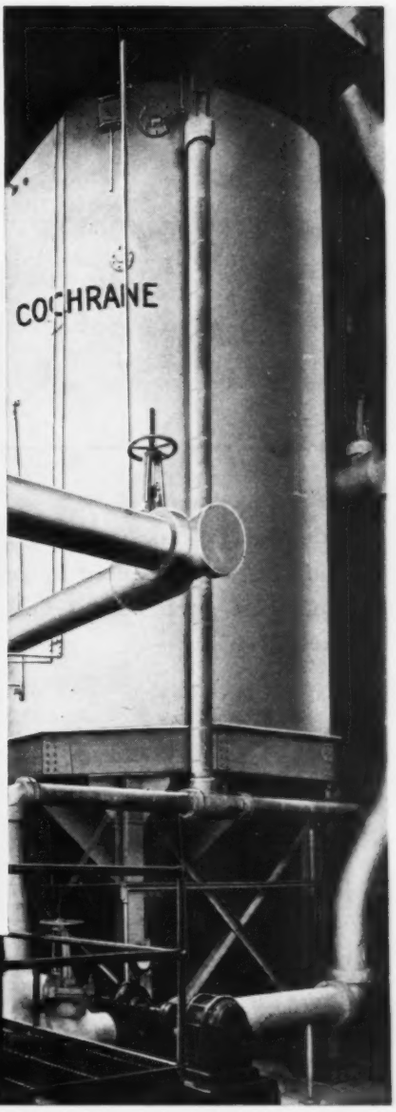
COCHRANE Feed Water Treatment prevents

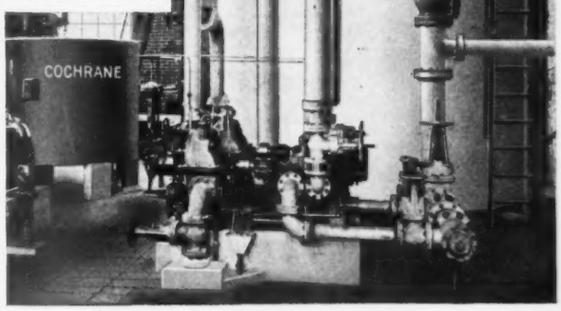
SCALE: by removing chiefly calcium and magnesium, and by giving the water such chemical characteristics that the remaining mineral solids do not form scale.

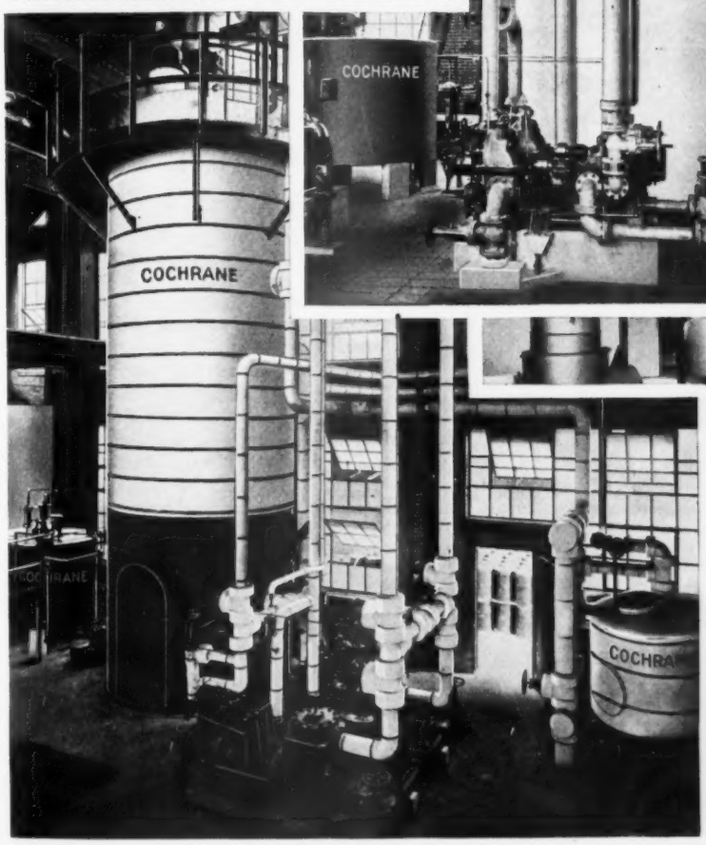
CORROSION: by removing oxygen from the water and establishing a proper pH value, or degree of alkalinity of boiler water.

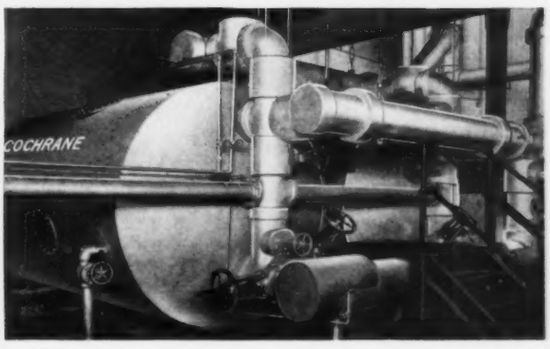
EMBRITTLEMENT: by maintaining a proper ratio of sodium sulphate to sodium carbonate, depending upon boiler pressure, as recommended by the A.S.M.E. Boiler Code Committee. The Cochrane Corporation pioneered the development of the Hot Process Softener and the Deaerating Heater, by which this is accomplished, and which can frequently be combined advantageously in a unitary structure. Let Cochrane's chemical and engineering staffs recommend the solution of your scale and corrosion problems.









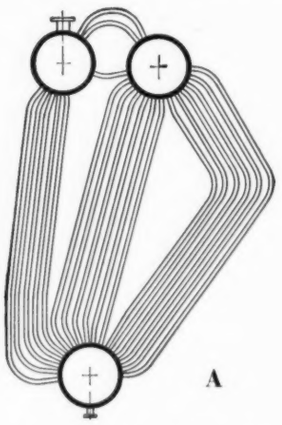


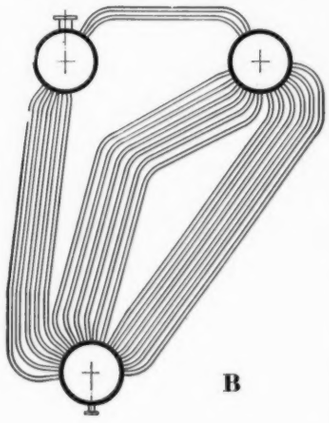
COCHRANE

WATER SOFTENERS • DEAERATING HEATERS • FILTERS • BLOW-DOWN EQUIPMENT • VALVES • FLOW METERS

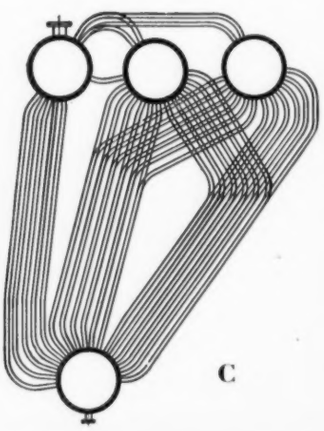
COCHRANE CORPORATION • Dept. D-11, 17th & Allegheny Ave., PHILADELPHIA, PA.

C-E BENT TUBE





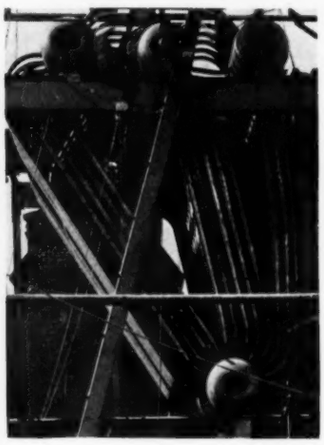
Superimpose A on B and you have the C-E Bent Tube Boiler, Design VA, shown below.

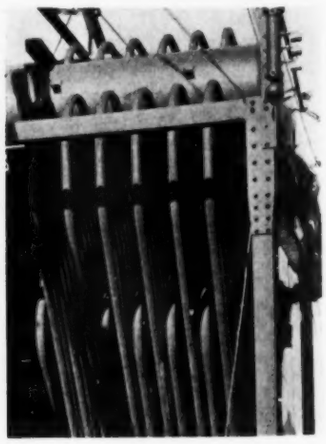


The Design VA Boiler has two separate circulatory systems. Drawings A and B may be considered as separate 3-drum boilers. When A is imposed upon B, the result is a 4-drum boiler with half the tubes comprising the first and second banks connecting to the first upper drum and the remaining half connecting to the middle upper drum. This equal distribution of the tube surface between the two drums is effected by having the first, third, fifth, etc. sections of tubes in the first and second banks connect to the front drum and the second, fourth, etc. sections in the same banks connect to the middle drum.

Introduced only six years ago, this boiler has received the endorsement, implied by purchase, of such companies as the following:—

ADOLPH COORS COMPANY
ALABAMA POWER CO.
CARNEGIE STEEL CO.
THE CHESAPEAKE CORPORATION
CHEVROLET MOTOR CO.
CLEARFIELD BITUMINOUS COAL CORP.
COLGATE-PALMOLIVE-PEET CO. (2 plants)
COLUMBUS RWY., PWR. & LT. CO.
CONTAINER CORP. OF AMERICA
DENVER TRAMWAY CORPORATION
CITY OF DETROIT, MICH.
DUPONT RAYON COMPANY
FORD MOTOR COMPANY (2 plants)
GARDNER RICHARDSON CO.
GLEN ALDEN COAL CO.
GULF REFINING CO.
CITY OF HASTINGS, NEB.
INDUSTRIAL COTTON MILLS CO., INC.
INTERNATIONAL HARVESTER CO.
INTERSTATE POWER CO.
JAMISON COAL & COKE CO.
JEDDO-HIGHLAND COAL CO.
THE ANDREW JERGENS CO.
KANSAS CITY WATER DEPT.
KENTUCKY-TENN. LT. & PWR. CO
OSCAR MAYER & CO., INC.
MICHIGAN PAPER CO.
MOORE STEAM TURBINE CORP.
JOHN MORRELL & CO.
NEW YORK CENTRAL R. R. CO.
PEERLESS WOOLEN MILLS
PENNSYLVANIA SALT MFG. CO.
CITY OF PIQUA, OHIO
PUBLIC SERVICE OF COLORADO
PULLMAN CAR & MFG. CO.
RATH PACKING COMPANY
THE RICHARDSON CO.
ROCHESTER GAS & ELEC. CO.
SYLVANIA INDUSTRIAL CORP.
TARENTUM PAPER MILLS, INC.
TENNESSEE PAPER MILLS
THERMOID RUBBER COMPANY
U. S. INDUSTRIAL ALCOHOL CO.
U. S. WAR DEPARTMENT
WESTINGHOUSE AIR BRAKE CO.
WESTINGHOUSE ELEC. & MFG. CO. (2 plants)
WILSON & COMPANY, INC.
CITY OF WYANDOTTE, MICH.





Side and front construction views showing the tube arrangement.

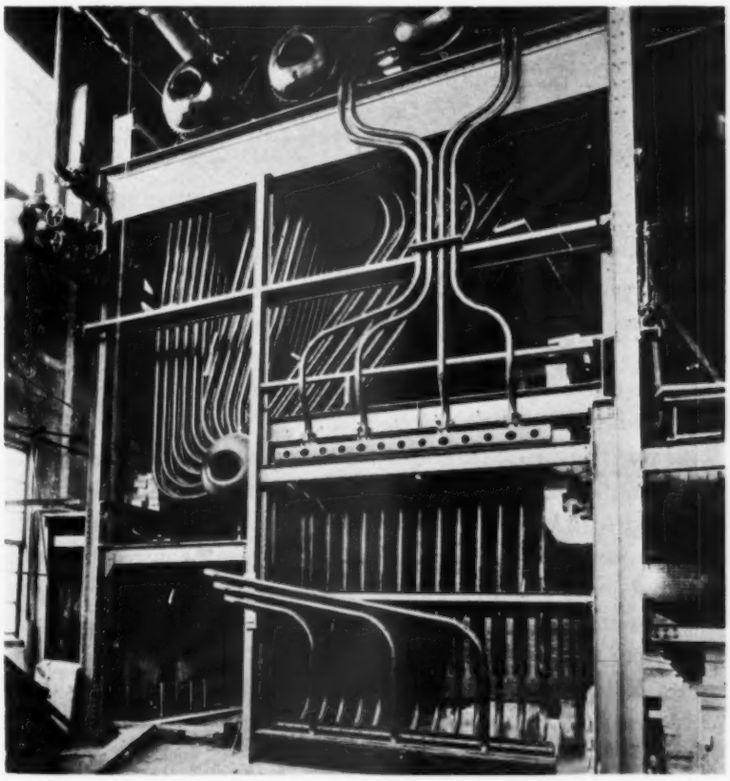
COMBUSTION

BOILER, design V-A

Six years ago, this advanced design of 4-drum boiler was placed on the market. Today it is installed in the plants of leading companies throughout the country—companies exceptionally qualified to determine the relative values of different boiler designs. Only marked superiority could account for such a record.

The design feature described and illustrated on the opposite page causes the discharge from the most active steam producing surface to be divided equally between two drums, with the result that both the steam liberating surface and the area available for moisture separation are virtually doubled. This in turn results in reduced turbulence and steam velocity and a nearly constant water level in both drums at all ratings.

By virtue of this exclusive design feature, the Design VA Boiler has established new performance standards for boilers of its class with respect to *rapidity of steaming*, *capacity per sq ft of heating surface* and *quality of steam*, especially at high rates of evaporation. . . . Combustion Engineering Company, Inc., 200 Madison Avenue, New York.



11,060-sq ft Design VA Boiler with C-E side and rear water-walls being installed in the Scotia, Cal., plant of the Pacific Lumber Company.

C-E PRODUCTS

BOILERS

Sectional Header
Box Header
Bent Tube
Fire Tube
Electric
Marine
(*Included in above list are all types known in the field as* **Heine, Walsh-Weidner, Casey-Hedges** *and* **Ladd**)

STOKERS

Multiple-Retort Underfeed
Single-Retort Underfeed
Type E
Type K
Stoker-Unit
Traveling Grate
Coxe
Chain Grate
Green Forced Draft
Green Natural Draft

PULVERIZED FUEL SYSTEMS

(*for boilers and industrial furnaces*)

Storage
Direct Fired
(*C-E Pulverized Fuel Systems, formerly known as* Lopulco, *include* Raymond *Pulverizers*)

FURNACES

Dry Bottom
Slagging
Water-Cooled Walls and Screens

COMPLETE UNITS

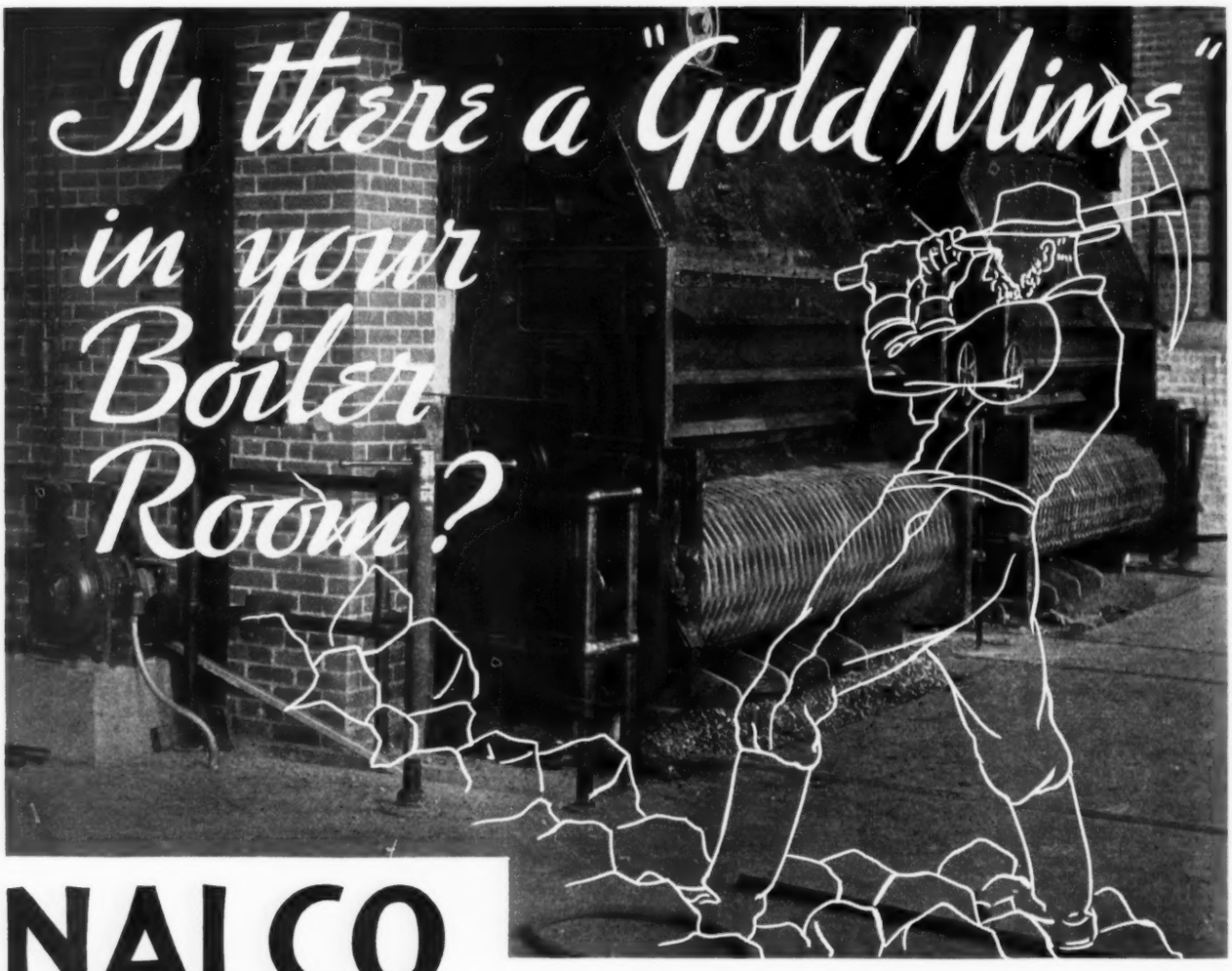
Units of standardized overall design
Coordinated designs comprising any combination of boiler and firing equipment

ALLIED EQUIPMENT

Air Heaters
(*Regenerative, Plate and Tubular*)
Economizers

A-111

ENGINEERING



NALCO

USER GETS $3,396 DIVIDENDS IN 11 MONTHS

PROSPECTING for a "gold mine" the Nalco way has a lot of advantages. Instead of climbing mountains and swinging a pick, you merely telephone or write Nalco headquarters and operations begin. With no cost or labor on your part, Nalco engineers make the survey and present you with a report showing just what dividends can be secured.

This was the procedure followed recently by the operator of an 800 h. p. mid-western boiler plant* and the net dividends amounted to $3,396 in the first 11 months. And it is significant to note that this saving was not a case of correcting any abnormal or unusual condition, rather, it was merely the normal result to be expected with efficient, effective water treating service: $2,291 less coal burned . . . Cost of chemicals (including Nalco Service) reduced $372 . . . Boiler cleaning costs $733 lower.

Nalco engineers are experts in finding these cost reducing opportunities. Ask them to make a survey in your plant. There is no obligation.

Name on request

NATIONAL ALUMINATE CORPORATION

6234 West 66th Place — Chicago, Illinois

EDITORIAL

Operating Procedure

When hand firing was general there were a few prescribed rules, modified to suit the kind of coal, which the fireman was taught would secure the best results. Beyond this his skill was largely a matter of experience and ability to make the most of his powers of observation.

The introduction of mechanical firing did not dispense with the need for observation and the ability to reason, but instead of a few set rules for firing it became necessary for the fireman to learn the operating procedure for the particular type and design of fuel-burning equipment, modified, in turn, by the fuel used.

When a stoker or pulverized coal equipment is installed the fireman is usually instructed by the manufacturer's representative in how best to obtain the desired results. But operating personnel changes, instruction books have a habit of finding their way into the front office for filing and, in some cases, operators lapse into indifferent habits or follow their own ideas as to how the equipment should be run. When the performance falls off it is but natural to blame the equipment. Sometimes this leads to a checking of the operating procedure by the manufacturer's representative, but more often the matter is not brought to his attention and the equipment continues to bear the onus. This is often the case when a change is made to coal having different characteristics from that originally burned.

In this connection it would be well to bear in mind that the manufacturer's service engineer usually has experience accumulated from many installations of the particular type of equipment, backed by the experience of those responsible for its design. This warrants consideration before departing from prescribed operating procedure. Moreover, if the prescribed procedure is followed and performance is not satisfactory the blame attaches to the equipment or to the engineer responsible for its selection, but if this procedure is not followed the responsibility for poor performance rests with the operator.

Dust Treated Coal

Bituminous coal, principally from the southern fields, is now being offered on the market after having been dust treated with either calcium chloride or oil. Generally, the coal is sprayed at the mine, although in some cases this is done at the yard as the coal comes off the belt conveyor. Both light and heavy oils have been employed and in either case a relatively small amount is used per ton of coal. This is sprayed in a fine mist which produces a very thin film over the surface of the coal particles. The treatment is especially applicable to slack.

Although this treated coal has been on the market for some time there appears to be confusion in the minds of some power engineers as to its purpose and scope.

In the first place, it should not be confused with colloidal fuel which is a mixture of oil and pulverized coal held in suspension. This offers some advantage as a marine or naval fuel, dependent upon a satisfactory fixative to provide stability to the mixture. Colloidal fuel was proposed and investigated in this country during the World War and it is again being experimented with in England.

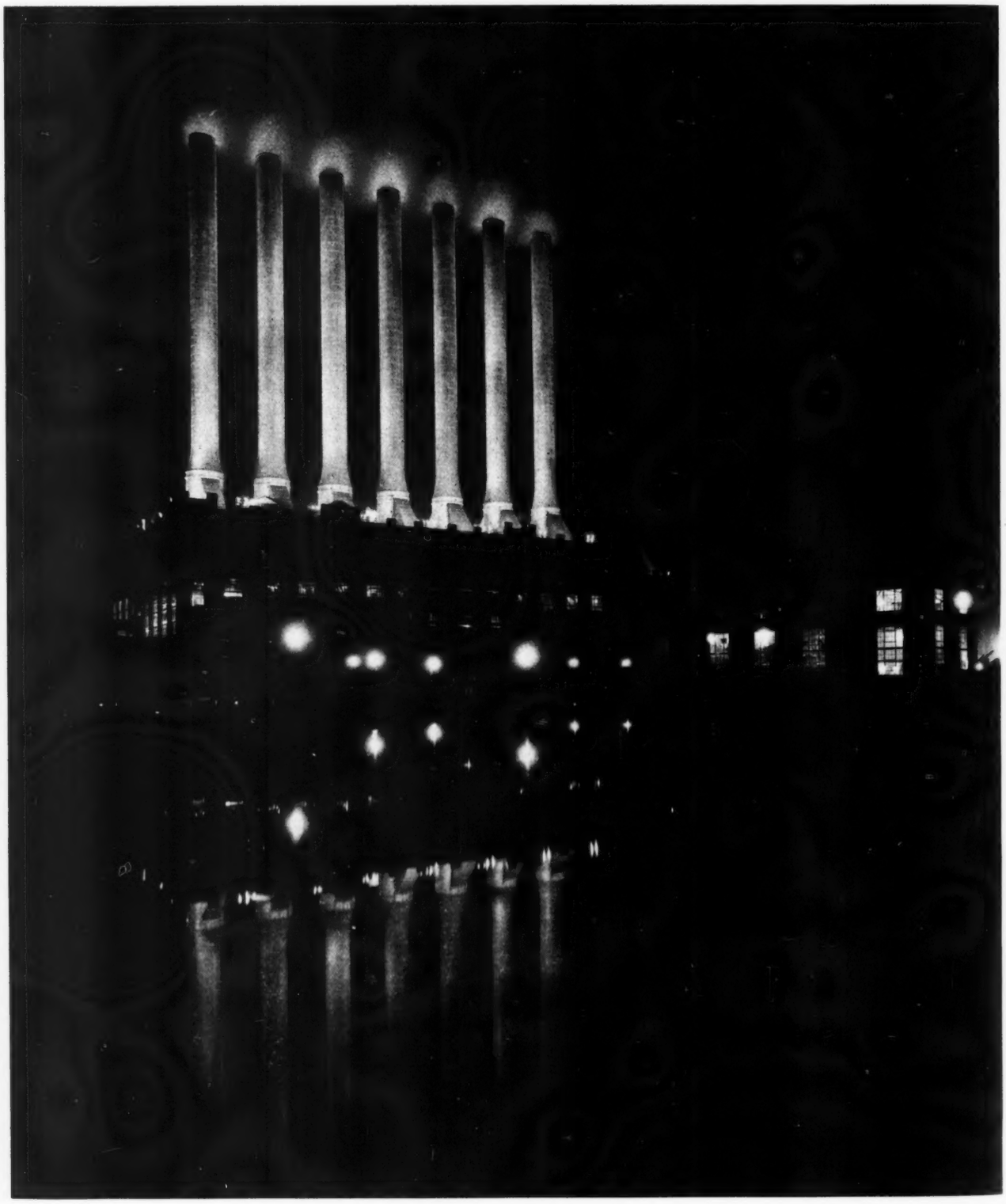
Dust treated coal, on the other hand, is distinctly what its name implies—it keeps down the dust in handling by conglomerating the fine particles and more or less coating the larger pieces. The quantity of oil is obviously too small to have an appreciable effect on the heating value or on the burning characteristics of the coal when hand fired or burned on underfeed stokers. In pulverized coal firing, however, the aim is to promote diffusion and suspension of the coal dust in the primary air stream—precisely what the oil in the treated coal is supposed to inhibit; hence there may be some question as to the effect of this dust treatment, if any, when the coal is to be burned in pulverized form. Data based on actual performance would be interesting.

In addition to keeping down the dust it is reasonable to assume that the oil would tend to protect the coal from moisture and oxidation and thus lessen the chance of heating in storage.

Many of the smaller plants are said to be using this treated coal, especially in cities where coal dust is objectionable, and one of the large utility plants is contemplating giving it a trial on underfeed stokers. In that case more factual information may be available.

The Story of Conners Creek

There appears in this issue the first of a series of six consecutive articles relating, in detail, the rebuilding of the Conners Creek Power Plant of the Detroit Edison Company. Several relatively short descriptions of this work have already appeared in the technical press, but it was the desire of COMBUSTION to present the complete story of this important undertaking. Descriptions are often interesting and have a reference or statistical value, but it was felt that to be of maximum value to the designing engineer the descriptive material should be accompanied by a discussion of the economics involved and the engineering thinking that preceded the various decisions. To this end the Detroit Edison Company cooperated heartily by assigning one of its engineers, Sabin Crocker, to prepare the material. Not only is Mr. Crocker well-known as an author of many technical papers but he has been closely associated with this rebuilding program since its inception and is therefore well qualified to write authoritatively on all phases of the work.



Conners Creek Power Plant, with its seven tall stacks, aluminum painted and floodlighted at night, has become a landmark for Detroiters as it is in plain sight from Belle Isle and much of the eastern part of the city.

Modernizing the CONNERS CREEK POWER PLANT

By SABIN CROCKER
Engineer, The Detroit Edison Company

This is the first of a series of articles relating in detail the rebuilding program at Conners Creek and discussing the various economic and engineering problems involved. The present article sets forth the disadvantages under which the twenty-year-old equipment labored and compares the new design features with the old, item by item. The discussion is confined largely to the first step in the rebuilding program, which embraces replacing the original three 20,000-kw turbine-generators with three 30,000-kw units and four of the old boilers with new boilers for 600-lb steam pressure at 850 F. Subsequent steps which will involve replacing the old 45,000-kw units with new 60,000-kw turbine-generators and the addition of more 60,000-kw units at the north end of the plant, are touched upon.

THE first section of the original Conners Creek Power House of The Detroit Edison Company went on the line April 26, 1915 as a plant of four boilers and two 20,000-kw turbine-generators operating at 220-lb steam pressure and 600 F total temperature. The power house as then projected was to consist ultimately of six 20,000-kw units served by twelve boilers, but subsequent developments in load growth and turbine design, coupled with the sudden expansion of war times, led to a change in plans. The original plant as actually completed consisted of three 20,000-kw, two 45,000-kw and one 30,000-kw turbine-generators served by fourteen boilers. The total installed capacity thus became 180,000 kw rather than 120,000 kw and the "firm" capacity of the Conners Creek load area, after deducting for running and standing reserve and crediting tie-line capacity from other plants, was reckoned at about 150,000 kw.

While the original plant was accorded some attention in its time as a well conceived project, ensuing years brought great advances in plant design through more efficient steam cycles and improved equipment. Consequently, as the 1930's rolled around, old Conners Creek did not compare so well with the Company's more recent plants or with modern practice elsewhere. The shortcomings consisted not only in a relatively high fuel consumption, but in an unfavorable comparison as to labor costs, ease in operation and the like. Neither were the old turbines, due to their antiquated design and lack of turning engines, so rugged and reliable as modern units nor as suitable for quick starting. As a result, they were ill fitted for peak-load operation to which service, as the least efficient units of the system, they should logically be relegated.

Furthermore, the same foundations, building space, condensers and auxiliaries, circulating-water canals, coal-handling equipment, stacks and the like could be salvaged and utilized to accommodate almost double the capacity in modern equipment with only slight alterations. The erection of a new and larger switching station separate from the old power house was imperative, a step which seemed called for anyway on account of the need for a different type of switch gear better adapted to the magnitude of electrical faults now possible in the transmission and distribution system remote from the plant itself. The plant as it now appears with the separate switching station known as Essex is shown in the photograph, Fig. 1. Essex switch house is the five-story building to the left and somewhat behind the power house.

Without attempting to draw a fine distinction between whether replacement was dictated by physical depreciation or by obsolescence through advancement in the art, the best business judgment of the management was for rebuilding now while load conditions permit having part of the equipment at a time out of service. In 1929 the Conners Creek area load reached 157,000 kw which exceeded the firm-capacity rating of the old station, as already defined. The plant serves a district on the east side of Detroit in which is concentrated much industrial load. Past experience, particularly that following the World War, has demonstrated that load recovery comes faster in this area than in the others, and that after a depression the comeback may be with startling suddenness. The Company's responsibility in serving its customers dictates that such a comeback be anticipated and provided for in advance.

In studying present and future possibilities for the Conners Creek site three proposals were considered:

1. Retain the old plant undisturbed for peak-load service and build a new plant alongside.
2. Superpose high-pressure equipment on the old.
3. Rebuild and extend the present plant.

The inherent advantages of the third scheme for the particular circumstances existing at Conners Creek are discussed in detail in this series of articles. Decision to rebuild and extend plant capacity to the limit of the existing boiler house, coal-handling equipment and canals will in no way interfere with the possibility of building a second power house adjacent to the present power plant if and when load growth warrants. Meanwhile the Company will benefit from the use of an efficient rebuilt plant which is simple to operate, well equipped with elevators and other conveniences, and better suited to the Detroit Edison system of plant load areas.

Building Structure

The final building structure of the old plant is shown in plan and sectional elevation in Figs. 2*a* and 3*a*. The plant is built on a site which in its natural state was a half-submerged marsh at the junction of Conners Creek and the Detroit River near the outflow of Lake St. Clair. Consequently, elaborate foundation work was required in the way of thousands of timber piles supporting a heavy concrete mat. This foundation has served its purpose admirably with only moderate settlements, and the ability to re-use both the foundation and superstructure is a major contribution to the rebuilding program. The layout as altered in rebuilding and shown in Figs. 2*b* and 3*b* is described in some detail below.

Boiler House

Recent advance in central-station design is exemplified in the much greater boiler capacity now obtainable within a given building space. Each of the fourteen old boilers was capable of continuously carrying 10,000 kw main-unit load plus auxiliaries with a possible output for short periods approaching 13,000 kw, thus matching very well both the firm capacity and maximum rating of the turbine-generator plant. Each of the new boilers going within the same building columns will be able to carry in excess of 30,000-kw main-unit load plus auxiliaries, giving a possible installed main-unit capacity of 390,000 kw if all fourteen boiler positions are used. While beyond the limitations of the old turbine house, this possible boiler capacity could be utilized to advantage by making a further addition to the turbine house at relatively small expense, which does not involve a corresponding outlay for water screens or canals.

Several factors contribute toward this spectacular increase to about three times the original boiler capacity within the given space, among which are:

1. The plant water rate has been decreased about 28 per cent while the heat added in the boiler room per pound of steam generated, despite the greater heat content per pound leaving the boiler, has been decreased about 5 per cent through the regenerative feed-heating cycle.
2. The economizers now installed above the boiler proper impart to the feedwater about 7 per cent of the heat added in the boiler room.
3. The provision of water walls has made possible a higher furnace temperature and a correspondingly greater rate of heat liberation per cubic foot of volume.
4. Metered combustion air control coupled with improved stoker design has made possible sustaining for long periods a much higher rate of combustion per square foot of grate area.
5. The addition of an air preheater has contributed toward carrying higher ratings, although its primary purpose is to obtain increased efficiency rather than greater capacity.
6. The development of a "compensating" superheater has made possible forcing the boiler to high ratings without an excessive rise in superheat.

In order to accommodate air preheaters as well as economizers and induced-draft fans in the new plant it became necessary to build a fan room on what was the roof of the old boiler house, as shown in Fig. 3*b*. The new brickwork also shows distinctly in Fig. 1. While space had fortunately been provided above the old boilers for economizers with their induced-draft fans, these were never installed and, at the time the plant was built, air preheaters were unthought of except possibly for some installations on shipboard. The forced-draft fans of the old plant were located in a fan gallery just below the

Fig. 1—Aerial view of Conners Creek Power Plant and Essex Switching Station

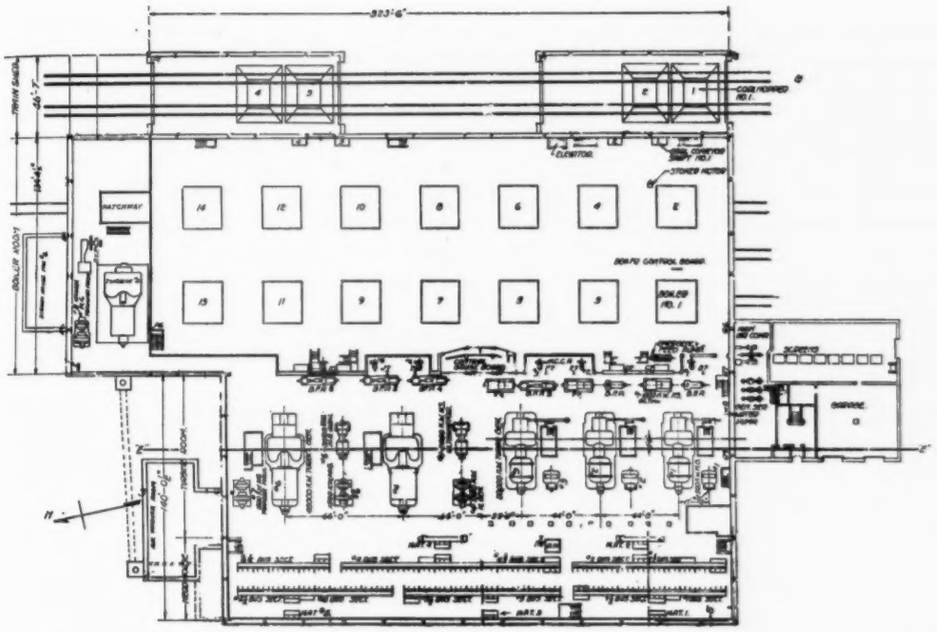


Fig. 2a—Plan of main floor, old plant

boiler operating floor, but with the advent of preheaters had to be replaced with new and larger fans located for convenience in the upper regions of the boiler house.

Turbine House

The plan of the old turbine room as shown in Fig. 2*a* requires some explanation. The arrangement of the three 20,000-kw units and their auxiliaries is that originally contemplated for the whole plant. Due to rapid load growth in turbine-generator size the next two units were stepped up to 45,000 kw, taking up the remaining space in the main turbine room. In the boom period immediately following the War, still more capacity was required in the Conners Creek load area and a 30,000-kw turbine-generator known as old Unit 8 was installed during 1920 in a separate, but connecting, turbine room occupying the space intended as an additional north bay of the boiler house. This completed the old plant. The installation of this 30,000-kw unit largely reflected a lack of confidence in the materials of the 45,000-kw Unit 4 which had been produced during the War. The original casing and head end of the No. 4 turbine were "semi-steel" (cast iron containing some steel scrap) instead of the cast steel which would normally have been used for superheated steam parts. The casing actually failed in service on July 16, 1920 before either the 30,000-kw unit or the second 45,000-kw unit were available for use. This failure, following one day after the third 20,000-kw unit had stripped its buckets, resulted in an awkward power shortage extending through several months while repairs and additions were being effected.

The original pumping arrangements and auxiliary

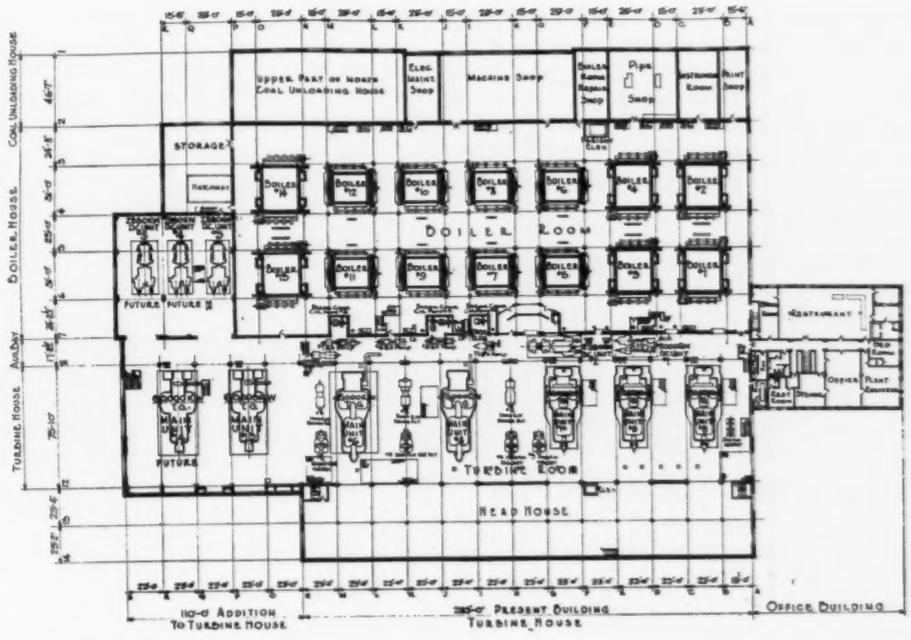


Fig. 2b—Plan of main floor, rebuilt plant

system were such as to require considerable space on the turbine-room floor. The auxiliary bay was crowded with boiler-feed pumps and house alternators, while the space between main units was taken up with exciters, motor-generator sets and more house alternators. The length of tubes used in the 45,000-kw condensers led to rather wide spacing between the 45's and between them and the adjacent 20, and this in turn fixed the placing of the present new units, to be known as Nos. 8 to 12 inclusive, to match the existing foundations and condensers. The closer spacing of the proposed 60,000-kw Units 13 and 14 in the turbine-house addition is made possible through the use of condenser tubes only 18 ft long instead of the 24-ft tubes used with the old 45,000-kw units.

The old 30,000-kw turbine-generator, No. 8, installed as an emergency measure after the War never was a very satisfactory unit to operate because its isolated position required additional attendance and involved an excessive pressure drop in the steam line. Hence, it was the first unit to be shut down during the depression, and no tears were shed when it was dismantled to provide a generator for the new high-pressure Unit 8. Its condenser and auxiliaries will be utilized in a new location for an additional 60,000-kw unit, and the separate turbine room, formerly housing the old 30,000-kw unit, will eventually become an auxiliary alcove for part of the new dc house-service units, with some space available for storage of spare parts.

In comparing the sectional elevations of the new and old turbine rooms it is apparent that only superficial changes were made in the building structure by way of removing the old roof monitors with their air ducts and exhauster fans. These were no longer required due to the relatively low moisture content of the atmosphere in the turbine room effected through an improved design of shaft packing installed some time ago on the old main units. The air-washer house at the north end of the old turbine house originally sheltered four large fans with vento-heaters and air washers, all of which were provided to supply conditioned air to the turbine-room basement from whence it was drawn through the generators for cooling purposes and discharged either into the turbine room or pipe gallery, eventually finding its way into the boiler house and the stoker blowers. This system was displaced several years ago through the installation of generator air coolers on the main units. One of the old fans was saved, however, and moved over with sufficient vento-heaters into the north end of the switch house basement where it still serves for year-round ventilation of the whole condenser room. The air-washer house itself was torn down.

The ability to continue using the old turbine-room crane and extend its travel to the addition at the north end was something of an advantage in itself. Fortunately none of the new turbine or generator parts weighs in excess of the crane's capacity, and its relatively infrequent use is conducive to long life.

Screen Houses and Canals

In extending the turbine house to the north to accommodate 60,000-kw Units 13 and 14 it was necessary to continue the mat and canal work under these units. This involved some alterations and additions to connect with the existing canals and to handle the increased water quantities. The main screen house of the old plant is located at its south end, while a smaller screen house which originally served old Unit 8 is located at the north end. Both screen houses will be utilized in the rebuilding scheme to supply circulating water from both ends of the plant into a common intake tunnel extending from north to south below the condenser room. The only changes contemplated are to cut a few holes through some concrete bulkheads underneath the plant so as to increase the canal cross-section at the points of maximum flow. Tests on canal models indicate that this additional flow area should afford sufficient head at the screens for passing the increased water quantity required to serve a total rebuilt-plant capacity of 330,000 to 390,000 kw.

The ability to carry double the plant capacity with only minor alterations in the water supply system is an outgrowth of the following developments which have taken place since the old plant was built.

1. The throttle steam rate is about 17 per cent lower for the new steam conditions.
2. The change from the Rankine cycle to the regenerative feed-heating cycle has transferred about 25 per cent of the main-condenser load to the feed heaters.
3. Rearrangement of tube spacing in the condensers has made it possible to carry maximum load at the same back pressure with about 10 per cent less water.
4. Chlorinating of the circulating water inaugurated in 1930 keeps the condenser tubes cleaner and is credited with maintaining a lower back pressure plus a reduction of about 10 per cent under the amount of water required for foul-tube conditions.

Items 1 to 3 inclusive are discussed in detail in subsequent sections of this article.

Switch House

As previously mentioned the old switch house, originally called the "head house," was already inadequate to hold the switch gear, transformers, regulators, etc., required to keep pace with the system-load growth and afford sufficient protection against the heavy current faults now possible. The interrupting capacity of the switches was less than the possible short-circuits, while sufficient segregation of equipment was impossible, and the cable vault was crowded and inaccessible. In general, since the electrical standards of this important station were considerably below modern requirements and below those of the rest of the Company's system, they constituted something of a hazard with continued operation of the old equipment. Since any increase in capacity on the Conners Creek site would immediately demand a revamping of the entire switching equipment, it was decided to proceed some time in advance with the construction of Essex Switching Station and let it serve the old generators pending rehabilitation and expansion of the power plant. Carrying out this intention Essex was placed in service in 1932, some two years in advance of the first rebuilt generating unit.

Essex as it now stands is built to accommodate 68 feeder positions, which should be adequate for a plant of at least 270,000 kw. When an aggregate of more generating capacity is installed, an extension can be made to the right end of the station if and as required. The transformers, regulators, switches, etc., are in the large main building, while the smaller house in front contains the control panel and operator's quarters. The outstanding features of electrical design may be summed up as follows:

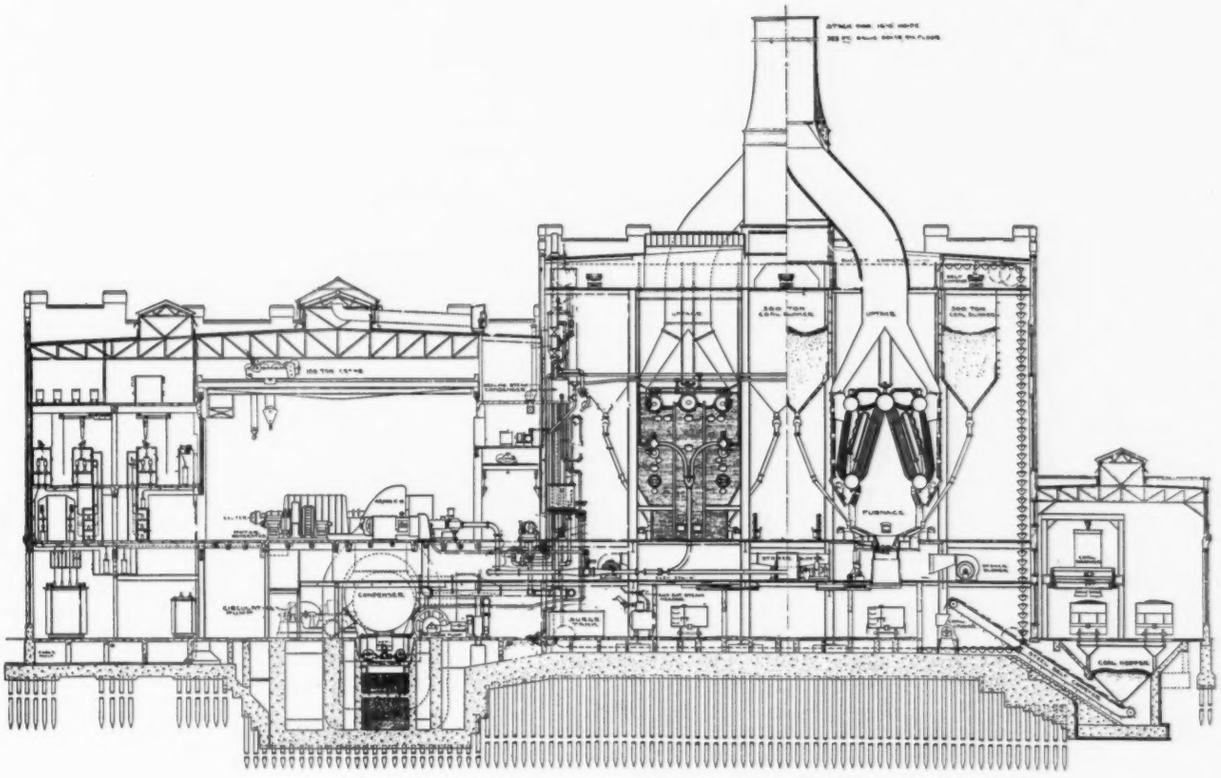


Fig. 3a—Typical cross-section, old plant

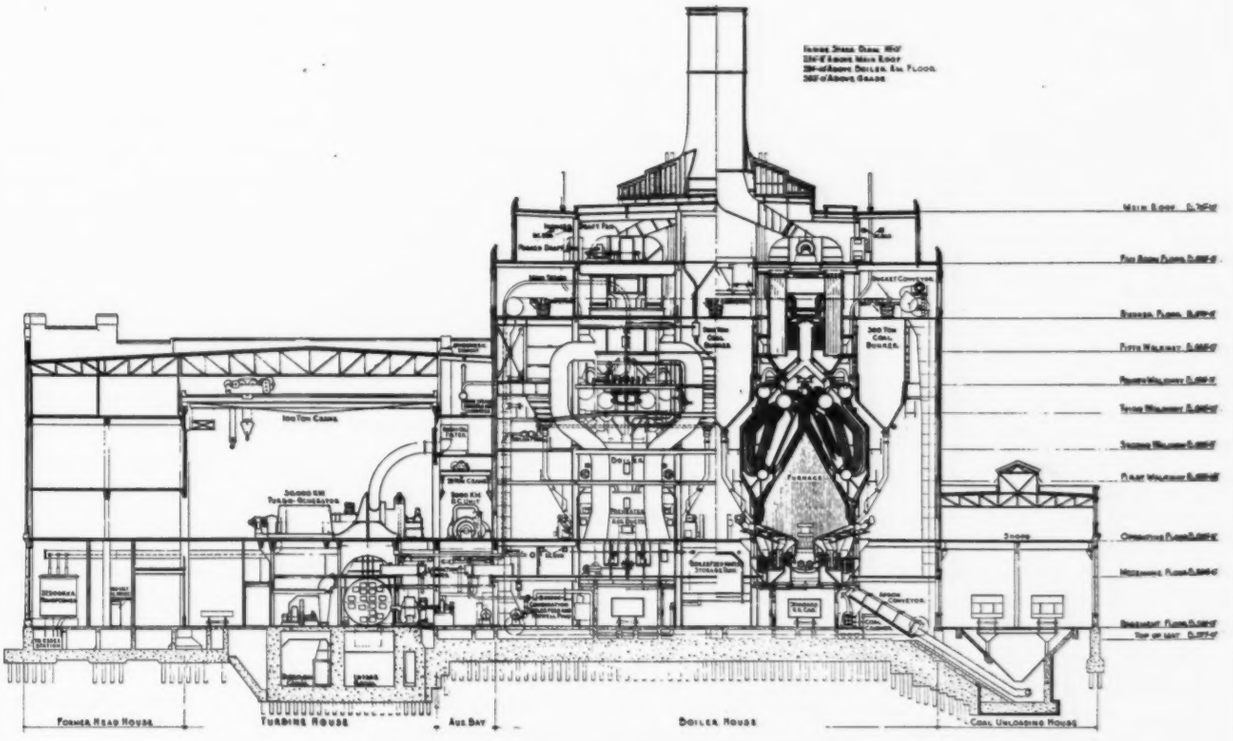


Fig. 3b—Typical cross-section, rebuilt plant

1. The interrupting duty of the circuit-breakers was kept within present-day ratings through a scheme of connections which provides fault-current limitation without handicapping power interchange between the Conners Creek Plant and the system as a whole.
2. Possible damage from explosions and ensuing oil fires was kept to a minimum by the use of H-breakers and oilless deion breakers, air-cooled reactors, and air insulated busses and by isolating groups of breakers in separate rooms. Where oil-filled equipment was used the oil content was kept to a minimum.
3. The clearing of all faults, including those directly on the bus, is made rapid and positive by inclosing all electrical equipment in isolated metal housings, and by the associated bus-fault relaying.
4. Fault currents for line-to-ground faults on the bus are kept to a predetermined, adequate path and not allowed to roam at will throughout the building. Two or three previous disagreeable experiences had shown that fault currents in the reinforcing rods could do considerable damage to the concrete at most unexpected places. Bus current faults are confined through the use of isolated metal housings and a low-impedance ground network of copper and building steel.
5. The electrical installation is divorced in so far as is possible from the building construction through the use of metal housings and armored control cable run in racks or chases instead of inside ducts buried in the concrete building structure. In this way the building was first erected like a warehouse into which the electrical equipment was later introduced.

The electrical features of Essex Switching Station were discussed in detail in a paper presented at the A.I.E.E. Winter Convention in January 1934.[1]

Prime Movers and Firm Capacity

In discussing prime movers, firm capacity and system ties an account of how the individual turbine-generator capacity was established should be of interest in bringing out the inter-relation of economics and engineering in the rebuilding project. For a plant ranging in ultimate size from 300,000 to 400,000 kw it would seem that turbine-generator capacities might lie between 50,000 and 90,000 kw. The choice was influenced in this case, however, by the ability to re-use certain equipment if the unit capacity were held down to suit. Another factor of prime importance was the effect of unit size on the firm-capacity rating of the station and the system. These considerations are discussed in some detail.

Among the major pieces of old equipment in first-class condition and available for re-use in the rebuilding project were two 30,000-kw 1800-rpm generators, one salvaged from the unit installed in an additional bay at the north end of the Conners Creek boiler house, and the other coming from a similar unit removed in dismantling old Delray Power House No. 2. The ability to place these generators with new 30,000-kw turbines on the centerlines of two of the old 20,000-kw, 1200-rpm units and thus re-use the condensers and auxiliaries practically undisturbed was a profitable item in the first step of the rebuilding project. The increase in speed from 1200 to 1800 rpm was an important factor in getting 50 per cent larger equipment in the same space, while adoption of the regenerative-feed-heating cycle with higher steam pressure and temperature coupled with improved tube-sheet layout were responsible for the increased turbine-generator capacity which could be served by the old condensers. Having replaced the first two of the old 20,000-kw units with 30,000-kw units, the next logical move was to buy one new similar 30,000-kw turbine-generator for replacing the third and last small unit. The first of the rebuilt 30,000-kw turbine-generators, No. 8, is viewed in Fig. 4 from the high-pressure end.

[1] See "Switching at the Conners Creek Plant," by A. P. Fugill, *Electrical Engineering*, January 1934, pp. 162–168.

Once thought out, this first step of the rebuilding project was predestined for execution. The best course to follow in succeeding phases was not so obvious. Several schemes were studied in connection with the old 45,000-kw units such as: (*a*) complete replacement of turbine-generators and auxiliaries with larger units of 60,000- to 90,000-kw capacity; (*b*) superimposing 15,000-kw high-pressure units exhausting to the old units as they stood; (*c*) buying new 60,000-kw turbine-generators to place on the old center lines, thus making it possible to re-use the old 45,000-kw condensers with their auxiliaries, practically undisturbed. The last scheme was found the most attractive from an all-round viewpoint since it got the most out of the old equipment without having to continue using obsolete turbines or resort to the awkward heat balance produced in this case by superimposing high-pressure elements. A single-barrel turbine was found more suitable than a multi-barrel unit of either the tandem or cross-compound varieties, from considerations both of cost and the ability to fit into the available space.

Since in the second step 60,000 kw was about the size limit for re-using the orientation and centering of the 45,000-kw units, the adoption of this scheme had a pronounced influence on unit capacity in the third phase of the rebuilding project which involved placing entirely new units in an addition at the north end of the old turbine house. Other factors being equal the preference naturally is in favor of having as many units identical as is possible.

Firm Capacity Influences Turbine-Generator Size

Considerations of firm capacity also had a marked influence on the economical choice of size for the new units. Company policy has established rules defining system and load-area firm capacity which take into account the possibility of outages in conjunction with relief from system ties and interconnections. The fact that turbine-generator output depends upon the availability of boilers and their capacity to produce the required amount of steam must also be considered. While it is normally expected to have sufficient boiler capacity to carry nameplate load on all turbines in the plant, this may be discounted somewhat where system-load conditions permit. The combination of these factors in reckoning firm capacity for the system and for the Conners Creek load area is shown in Tables I and II, respectively. While generator overload capacity is available for extreme emergency in some instances, no credit is taken for this in computing nominal firm capacity since ability to use it depends upon several uncontrollable variables such as additional margin of boiler capacity, low-temperature of circulating water, etc.

Definition of Firm Capacity

The basis for reckoning nominal firm capacity of the system for design purposes is intended to balance the need for continuity of service against the likelihood of simultaneous outages of units during periods of high system load, taking into account probable reliability of equipment as gathered from previous operating experience. Obviously, consideration of the capital charges incurred for spare units will tend to discourage extravagant assumptions as to outages. The Company's practice is to define system firm capacity as the sum of the rated generator output at all plants, modified if necessary by

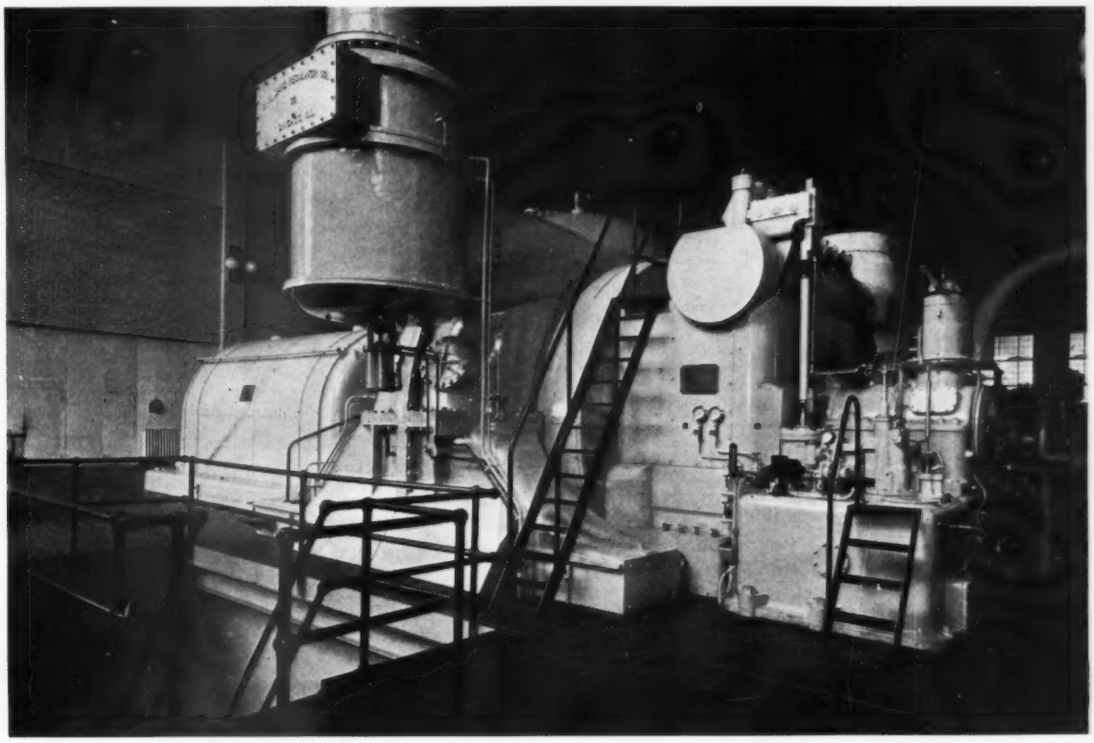


Fig. 4—No. 8 30,000-kw turbine-generator viewed from high-pressure end

boiler capacity, plus aid from other sources, less the rated output of the two largest units on the system. Aggregate generator capacity is designed to keep firm capacity as nearly as feasible just above maximum system load.

This aspect of firm capacity has its practical application in the following manner. The period of maximum load is considered as extending through the season of short daylight hours, from about October to March, during which relatively large lighting and power loads overlap in the early morning and late afternoon. Some shifting of this peak-load season takes place, of course, due to unusual spurts of industrial activity coming during other months. The coincidence of low solar illumination due to severe storm conditions with an hour of unusually high industrial activity is apt to result in a peak load either in summer or winter. Before the period of high loads, which past records indicate are more apt to occur during autumn and winter, all pre-scheduled and routine repairs are completed. Although every unit is then considered to be in good condition, account is taken of the possibility of unexpected repairs preventing, as a severe assumption, running of the largest unit, plus a further emergency outage of the next largest machine during the critical period. Power aid up to 25,000 kw also is available by prearrangement over tie lines from neighboring systems, industrial interconnections and hydro stations. The general system diagram of Fig. 5 shows the relation of Conners Creek to the rest of the system, the installed generator capacity at each of the four steam generating plants, and the interconnection between them.

Table I illustrates the system firm capacity as just outlined for two alternative asumptions. As the proposed 60,000-kw, or the 90,000-kw units considered at one time, would be the largest connected to the system, two of these machines are indicated as being on emergency outage. This is an extreme condition and possibly not a likely one, but being so chosen, it covers all other combinations of units such as a 30,000-kw and a 50,000-kw, and so on. Paradoxically, if the Company's firm capacity rule is followed, the change from moderate size units to a considerably larger size does not bring about a corresponding increase in firm capacity, since the addition of two new units of greater capacity than any previously installed will add firm capacity equal only to that of the two previous large units.

A general rule for running reserve covering all-year operation is derived from and is complementary to the definition of system firm capacity. It is the intention to have sufficient units running on the system at all times so that the load can still be carried with assistance from outside sources if the largest unit should trip out. Sustained outside assistance of from 10,000 to 25,000 kw from interconnections is arranged for in the event of emergency but is seldom utilized.

The power-plant-load-area firm capacity can be comprehended by visualizing a load area as a small separate system subject to the same criterion as the major system. Each power plant with the aid of tie-lines must be capable of supplying its load area. Load-area firm capacity then is the sum of the generator ratings, subject to boiler-output limitations, less the capacity of the two largest units, plus tie-line aid from the system. Table II illustrates this with the Conners Creek plant as an example having 30,000-kw units plus either 60,000-kw or 90,000-kw units. Note the assumption that two generators may be shut down in any given plant does not contemplate that a similar condition will exist simultaneously in any other plant.

These considerations influenced the decision on generator capacity at Conners Creek. Having arrived at a satisfactory program for the first step in rehabilitation by rebuilding the 30,000-kw units, the management com-

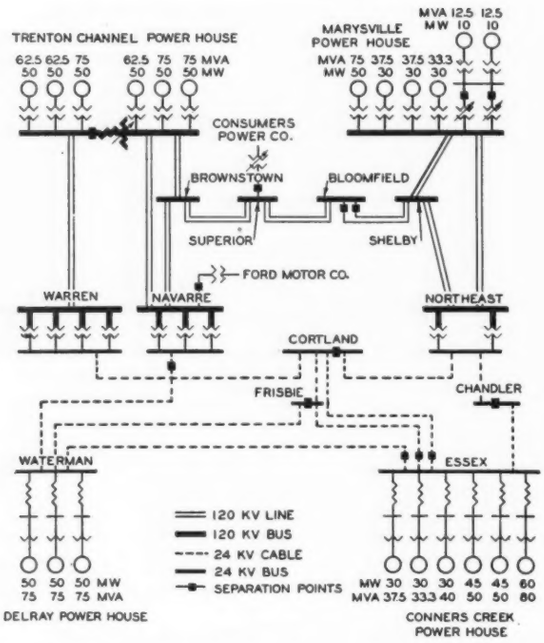


Fig. 5—Diagram of Detroit Edison electrical system

pared the installation of either 60,000- or 90,000-kw units. Among several reasons for favoring the 60,000-kw units was the consideration of system and load-area firm capacity. As shown in Tables I and II, the use of 90,000-kw units produces 30,000 kw less of firm capacity (according to the definition) despite 30,000 kw more of nameplate rating. This is due to the fact that in the Detroit Edison system, where the largest units have been 50,000 kw, the addition of the first two 90,000-kw or the first two 60,000-kw units would establish an increase of only 100 kw of system firm capacity. With the addition of four 60,000-kw units, however, the increase in firm capacity would be 50 + 50 + 60 + 60 = 220 mw, or 0.917 kw of firm capacity per kilowatt of installed system capacity. With three 90,000-kw units, the increase would be 50 + 50 + 90 = 100 mw or a ratio of only 0.704.

The impracticability of duplicating turbines already installed at other plants due to the higher pressure and temperature to be used, the possibility of re-using old auxiliary equipment, considerations of load growth and firm capacity and the approximately equal cost of either size of new equipment per installed kilowatt all indicated the most economical addition to plant capacity to be by single-barreled 60,000-kw turbine-generators.

System Flexible for Area and Base Loading

For many years it had been the practice to make each power plant carry its share of the load at all hours of the day. Ties were installed to provide relief from the system in case of emergency, but under normal operation power was not transferred to any great extent between load areas. Reduced loads during the depression, however, suggested the possibility of using tie lines for considerable transfer of power between plant areas, thus effecting substantial fuel saving. After extensive studies of plant heat rates and transmission losses, economies were effected, starting in May 1933 by a three-fold change in program consisting of carrying a minimum base load or more strictly a block load on the least efficient stoker-fired plant (old Conners Creek), giving large block loads to the more efficient stoker-fired plants (Delray and Marysville), and taking load swings on the more flexible pulverized-coal plant (Trenton Channel). Because of the ties already installed, an economical loading schedule could be followed without an attendant addition of new tie lines. Not long after block-loading was put into effect, some desirable changes in the layout of the Delray-Conners Creek tie cables were made, and two additional cables, made up of salvage and stock lengths, were laid. While not altogether necessary at the time, these cables are good insurance for continuity of service while certain units are unavailable during the rebuilding project. Tie-line cable entering Essex Station permit an exchange of power up to 70,000 kw. A partial view is shown in Fig. 6 of the control room of Essex which is the control and switching station for the Conners Creek generating plant and tie-line interconnections. The Conners Creek area is thus adequately safeguarded for an emergency either in the event of a return to area loading, or with block loading as at present.

TABLE I

Showing Effect of Unit Size on

FIRM CAPACITY OF SYSTEM

in Megawatts

	Three 30,000-Kw and Four 60,000-Kw Units at Conners Creek			Three 30,000-Kw and Three 90,000-Kw Units at Conners Creek		
Power Plant	Installed Generator Capacity	Firm Boiler Capacity	Output Available for System Firm Capacity	Installed Generator Capacity	Firm Boiler Capacity	Output Available for System Firm Capacity
Trenton Channel	300	300	300	300	300	300
Delray No. 3	160	153	153	160	153	153
Marysville	160	165	160	160	165	160
Conners Creek	330	330	210	360	363	180
Other Sources			25			25
System Total	950			980		
System Firm Capacity			848			818

NOTE: All units on system available except two largest turbine-generators on emergency outage.

Fig. 6—Control room, Essex Switching Station

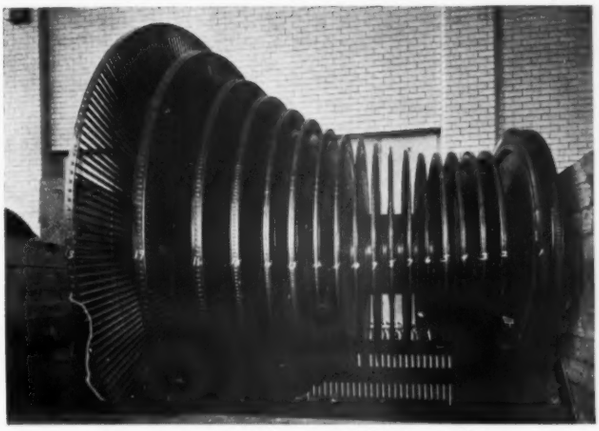


Fig. 7—Rotor of No. 8 30,000-kw turbine

The New Turbines

The turbine room of the rebuilt plant together with the turbine-room extension, as already described, will house three 30,000-kw and four or five 60,000-kw main units. As a detailed discussion of the heat cycle of these units will be given under "Heat Balance" in a later section, this description is confined to considerations of turbine construction and material.

The turbine rotor of the 30,000-kw machines, illustrated in Fig. 7 consists of one two-row Curtis wheel and seventeen single-row Rateau-type wheels. Steam for feedwater heating is bled at four stages, the 4th, 8th, 11th and 14th. The 60,000-kw turbine rotor consists of one double-row Curtis wheel and thirteen single-row Rateau-type wheels. Steam is extracted at three points for feedwater heating the 4th, 8th and 11th stages. Unlike the old units in which the number of stages increased with capacity of the turbine-generator, the new large unit actually has less stages than the smaller unit. Economic studies of alternate proposals for 17- or 14-stage 60,000-kw turbines based upon an annual plant factor of about 45 per cent (a conservative figure derived from Trenton Channel records) indicated that the annual heat saving by the 17-stage unit was insufficient to amortize the increased investment for the three extra stages. Further studies indicated a similar situation in the case of three versus four stages of feedwater heating, which will be discussed fully in the "Heat-Balance" section.

A turning motor, mounted on the middle turbine bearing of each new unit and geared to the shaft with an automatic release, revolves the rotor at $1^1/_2$ rpm after the unit goes off the line. As a result the rotor temperature is equalized so that the shaft is kept nearly straight at all times during the cooling cycle and hence in a better condition for quickly putting the machine on the line again when needed. An attendant result of a straighter shaft is decreased rubbing of the high-pressure and diaphragm packings, permitting closer clearances with less leakage and accordingly higher efficiency. Use of the turning motor almost entirely eliminates the uncertainty of being able to bring a machine up to speed without rubbing off the packing, and this in turn reduces the time required to put a machine on the line.

Before proceeding to a general description of the new units mention should be made of the parts salvaged from the old 20's and 30's and re-used on the new Nos. 8, 9 and 10 units of 30,000 kw. All three new units include the exhaust hoodsof the old 20's; other salvaged parts from the two old 30's were available for new units 8 and 9. These included the operating and emergency governor, the turbine base, the oil tank and oil pumps, the turbine middle bearing and oil deflectors, the coupling and the generator. No salvaged parts were used for the new 60's aside from the condensers and their auxiliaries.

Built within the last two or three years for the same steam conditions there is naturally great similarity in the materials and construction of the 30,000-kw and 60,000-kw turbines. Substantial differences exist only in the extent to which each material is used to suit changes in temperature range incidental to 14- versus 18-stage expansion through the turbines. Each unit is of the partial-admission type with eight control valves admitting steam to the first-stage nozzles. In the 30's all valves are located on the top, but for the 60's only four valves are located in the top of the high-pressure casing, the other four being in the bottom. This latter arrangement is intended to give very uniform heating of the upper and lower halves of the casing, thus minimizing expansion stresses. All valves are under control of the operating governor. There is no overload valve supplying steam at some stage other than the first as was the case in the 50,000-kw 1200-rpm turbines of the Company's system.

The control-valve casing and first-stage-nozzle chest is a separate casting of manganese-molybdenum steel inserted into the high-pressure casing of cast carbon steel. The throttle-valve body also is a manganese-molybdenum-

TABLE II

Showing Effect of Unit Size on

FIRM CAPACITY OF A LOAD AREA

in Megawatts

No. of Units Installed			Firm Boiler Capacity	Installed Generator Capacity	Two Largest Units	Available Plant Output	Tie Line Aid	Firm Capacity
Thirties	Sixties	Nineties						
3	4		330	330	120	210	70	280
3		3	363	360	180	180	70	250

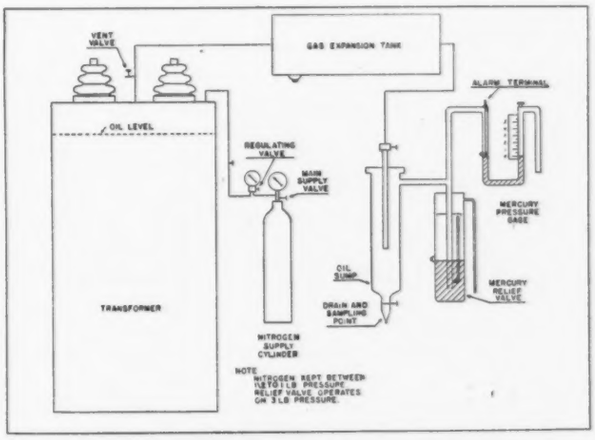


Fig. 8—Nitrogen equipment for transformers

steel casting. Machine steel is used for the control valve disk and seats, while the stems and bushings are Nitralloy. The turbine rotor consists of a forged carbon-steel shaft with turbine wheels shrunk and keyed in position. Different steels, namely, nickel-chrome-molybdenum steel, carbon steel and nickel steel, are used for the various wheels as steam conditions require. The turbine buckets are of a non-corrodible chrome-iron alloy throughout. In addition, the last-stage buckets are fitted with Stellite protecting strips. The upper stage diaphragms are of steel with welded-in nozzles, while cast-iron diaphragms are used in the lower stages. The exhaust hood is made of cast iron.

Operation of the 30,000-kw turbine-generators, sucessfully graduated from the usual minor troubles of the starting-up stage, now includes a high percentage of generation at full capacity, and gives every indication and expectation of continued satisfaction.

TABLE III
COMPARISON OF OLD AND NEW 3-PHASE 60-CYCLE GENERATORS

	Rebuilt Plant				Old Plant		
	Reconstructed Units		New Units		Old Units		
Unit Number	8	9	10	11, 12, 13	1, 2, 3	4, 6	8
Generator capacity, kw	30,000	30,000	30,000	60,000	20,000	45,000	30,000
Power factor	0.80	0.90	0.75	0.75	0.80	0.90	0.80
Continuous output, kva	37,500	33,333	40,000	80,000	25,000	50,000	37,500
Terminal voltage	12,200	12,200	12,200	14,400	4,600	12,200	12,200
Poles	4	4	4	4	6	6	4
Speed, rpm	1800	1800	1800	1800	1200	1200	1800
Closed air circulation, cfm	60,000	60,000	70,000	110,000	50,000	85,000	60,000
Exciter voltage	250	250	250	250	250	250	250
Max field current, amps	575	450	535	715	465	585	575
Gen. efficiency at full load, %	97.3	97.2	97.4	97.9	96.3	97.1	97.3

Generator-Design Improvements

The design of the generators represents the incorporation of many improvements to increase efficiency and to reduce temperature in the windings. Before enumerating the design features of the new units, some mention of the history of the re-used generators will be given.

Those used for the rebuilt units Nos. 8 and 9 when first installed in 1920 as old No. 8 Conners Creek and old No. 10 Delray respectively, were equipped with open ventilation. After a few years of operation the insulation failed, and both generators which had originally the old-type coil winding, had in turn to be re-insulated. Between the time that old No. 10 was rewound and old No. 8 failed, the transposed or bar winding was introduced and advantage of this was taken to increase the efficiency and kva rating of old No. 8 generator. This is now new No. 8 as shown in Table III along with data on the other generators. While these repairs were being effected, air coolers and closed-circulation systems were installed. The rotors of each machine were also circumferentially grooved to provide more effective cooling surface. No changes in either generator were necessary at the time they became part of the present rebuilt units.

With the installation of No. 10 unit which partially duplicates Nos. 8 and 9, non-magnetic-steel end-rings were introduced into the Conners Creek plant. Their desirability lies in the minimizing of heating due to induction losses in the ends of the generator. Solid coupling of the turbine and generator shafts has been adopted for the first time in the Company's history for large units with four main bearings.

The generator rotor of No. 13 unit is a three-piece nickel-steel forging with axial and radial holes to distribute more evenly the air cooling. The frame is of built-up steel-plate welded construction with provision for multiple-path ventilation. The increased terminal voltage indicated in Table III has effected reductions in the cost of generator leads and the auto-transformer. Departure from previous design is witnessed in the introduction into the closed-circulation air-cooling system of four external motor-driven fans, which are estimated to be 50 per cent more efficient than the integral fans of all previously-installed generators. Three motors are direct-connected through transformers to the generator; the fourth is supplied from the house service for use during starting periods and for periods of high circulating-water temperature. Each has individual full-voltage starting at 230 volts. These separate controls allow power savings at part load and with low water temperatures. The closed circulation of air is utilized for the first time in cooling the overhung totally inclosed exciter and the field collector-rings. A filter is provided to eliminate any dust particles that are produced at the carbon brushes.

Transformer Design Improved

No story of the Conners Creek generators would be complete without mention of the auto-transformers which connect the generators with the 24,000-volt bus at Essex.

By using the old 30,000-kw generators, it was possible to re-use the old 37,500-kva auto-transformers. Before being put into service, these were thoroughly cleaned and the tanks hermetically sealed to withstand a nitrogen pressure of 7-lb gage. Nitrogen is supplied above the oil level at $^1/_2$ to 1 lb through an automatic pressure-reducing valve from tanks at 2000 lb. The gas pressure is prevented from exceeding 3 lb during loading and unloading cycles of the transformer by a non-return pressure-relief valve discharging to atmosphere, any loss of gas being made up anew from the supply tank. As an additional protection each transformer tank has a spring-loaded pressure-relief diaphragm. A diagram of the equipment is shown in Fig. 8.

While the absence of oxygen eliminates the tendency of the oil to oxidize or sludge the primary purpose is to prevent explosions. Both No. 10 40,000-kva and No. 13 80,000-kva transformers are to be similarly equipped with inert gas. The same principle also will be carried out for the totally inclosed high-tensions potheads on these two units. All transformers, some of which are located in existing indoor cubicles, and some outdoors, are water cooled with general-service water which also supplies fire-fighting equipment.

An account of the novel features of the new Conners Creek stokers with their improved design of combustion air control will appear in the December issue. Reasons for installing a new set of stokers rather than switching to pulverized coal burning are reviewed, and the development steps in perfecting the present metered air control are described.

DISTRICT STEAM HEATING-III

Today and Tomorrow

By ROGER D. DeWOLF
Consulting Engineer,
Rochester, N. Y.

The previous articles, which appeared in the September and October issues, reviewed the handicaps under which district heating started, its growth and development, its present status and its general relation to the electric utility. The present article, which concludes the series, looks to the future and points out the economies of higher pressures and temperatures, both in steam generation and in distribution. It also shows the savings in peak capacity cost, as well as greater thermal efficiency attainable, for a utility supplying both steam and electricity, where part of the electric load is supplied from automatically operated back-pressure turbines located at strategic points on the steam distribution system.

IN the early days gross income received by the utility through the sale of electricity was such that, even though income from the sale of steam was disproportionately small, the total from electricity plus steam was sufficient to give an attractive return on the combined investment. Drastic reduction in electric rates during the past twenty-five years, together with a remarkable increase in the efficiency of customer use of electricity, has resulted in a reduction of the gross income received from the sale of electricity and at present the utility that furnishes a combined service receives from the average customer buying this combined service, more gross income from the sale of steam than from the sale of electricity.

As has been previously pointed out, much of the steam now sold is being furnished to customers in the competitive class. In meeting the customer's own costs in this competitive field, it behooves the utility to take advantage of all the economies of combined production of steam and electricity possessed by the customer's own plant, as well as such other advantages which the utility possesses due to its combination of the requirements of many customers, large scale production, distribution, etc. Technical development in the past ten to fifteen years in power-plant equipment and in the field of steam distribution has put the utility in a far more favorable position to render a combined steam and electric service than was the case in the past. It is likewise true that the development of power-plant equipment has put the isolated plant supplying electricity and steam for space heating and process work, in a far more favorable position especially to meet competition of the utility that supplies electricity only.

It seems to the writer that these developments enable the utility to meet more effectively competition from the isolated plant, by furnishing a combined service, which completely, not partially, replaces the service rendered by the latter; to cooperate with a customer in improving the efficiency with which he uses both electricity and steam; and in the coordination of these demands.

To provide the necessary facilities to render such a combined service more generally available, will require a considerable development of steam distributing systems. Certain companies started such a development just prior to or during the depression period. The writer discussed this subject recently with the chief engineer of a company which spent several million dollars on its steam business between 1930 and 1934. He expressed the opinion that under present-day conditions, the steam business offers a better field for development, taking into consideration its coordinating value with the sale of electricity, than does the electric business alone. This same engineer, ten years ago, was markedly apathetic to the steam business.

The principal factors which place the steam business in a much more favorable position for development than at any time in the past are:

1. Development of high-pressure boiler plant equipment which is cheaper, more efficient and capable of carrying higher sustained overloads than corresponding equipment of ten years ago.
2. Development of new equipment and construction methods for higher pressure steam distribution systems. Both electric and gas welding have been developed to the point where there is no further question as to their reliability. Line losses have been greatly reduced; more efficient insulating methods developed; dry steam in the line under almost all conditions has practically eliminated water hammer.
3. Metering equipment has been improved and cheapened; automatic temperature control devices have been perfected to markedly reduce the steam consumption; and more efficient heating equipment has been introduced.
4. More study has been given to the coordination of steam and electric demands; to the reduction in capital expenditure and operating costs by supplying a portion of the electric demand by means of non-condensing turbine-generators acting as reducing valves to supply low-pressure steam for space heating and process work.

Boiler Plant Equipment

With the improvement in boilers, furnaces, water walls, stokers and powdered fuel-burning equipment now available, the cost per 1000 lb per hr of steam generating capacity of a plant designed for 400-lb pressure, may be less than that of a plant for 200 lb. We can, in effect, regard 400 to 500-lb operating pressure as the present-day standard. Boilers of this order of pressure have been in continuous service now for ten- to fifteen-

year periods. We have learned how to handle them; have learned how to cope with the feedwater problems and, in general, have found them to be more reliable than the lower pressure boilers. The piping, valves and other accessories are all smaller, more rugged and less subject to outage.

Improved operating methods have raised operating efficiencies and brought them closer to test efficiencies. Systematic inspection has reduced maintenance and practically eliminated unexpected outages. We have found by actual experience that 400-lb equipment is safer to operate, cheaper to maintain and more reliable than our older 200-lb plants.

The high steaming capacity of the modern high-pressure boiler unit, per cubic foot of space occupied, and particularly per square foot of floor space, has reduced building and real estate costs. Also, its overload capacity has reduced the number of standby banked boilers formerly considered necessary.

Automatic combustion control maintains efficient operation with either stokers or powdered fuel. Monthly operating efficiencies averaging from 78 to 86 per cent are now attainable under normal conditions, and higher efficiencies have been attained in some plants designed particularly for high efficiency and operating under favorable load conditions. Both stokers and powdered fuel equipment are today handling fuels formerly regarded as difficult to burn efficiently.

Distribution System Equipment

Modern welding has revolutionized this field in the past fifteen years. The elimination of practically all gasketed joints has greatly reduced the unaccounted-for line losses. Expansion joints of improved types take care of the expansion of 150 to 200 ft of pipe, so that manholes can be located 300 to 400 ft apart, with anchor points between. Creased bends give greater flexibility and can be used to advantage in certain cases. Many improvements have been made in underground construction and insulation, particularly for higher pressures and temperatures. Automatic pressure-reducing valves and desuperheating equipment have been improved so that any desired pressure and temperature conditions can be maintained at any specified point in the distribution system.

With steam lines capable of operation at 400 to 500-lb pressure and temperatures of 500 to 600 F available, the pipes required undergo a remarkable reduction in size. Where formerly good judgment dictated a maximum velocity when transmitting dry, saturated steam (more saturated than dry, as a rule), of 6000 ft per min, today with moderately superheated steam one can safely figure on 12,000 ft and for peak loads, 15,000 to 18,000 ft per min.

With the interior surface of the pipe dry, and with superheated steam, the heat loss per square foot per degree difference per hour is only about one-half that obtaining when moist steam is being transmitted. With the square feet of pipe surface greatly reduced, due to smaller pipe, distribution system losses become very low. In fact, there are lines in which, over a period of years, by maintaining a moderate degree of superheat, the pounds of steam delivered equalled the pounds of steam supplied over monthly periods. Theoretically, this should be the case, since all the heat lost by radiation from the line, is supplied by the drop in temperature of the steam between the entering and the discharge ends.

To indicate the comparative pipe sizes for low-pressure saturated steam as compared with high-pressure superheated steam, let us assume the following conditions:

Steam transmitted, 150,000 lb per hr.
Saturated steam at 50, 100, 150, 200-lb gage.
Superheated steam at 300, 350, 400 and 450-lb gage.
Superheat 50 deg.

$$\text{Lb per hr} = \frac{A \times V}{sv} \times 60$$

$$\text{or } A = \frac{\text{lb per hr}}{60V} \times sv$$

A = cross-sectional area of pipe in sq ft.
V = steam velocity in ft per min.
= 6000 for saturated and 12,000 for superheated steam.
sv = specific volume of steam.

$\frac{\text{Lb per hr}}{60V}$ = 0.4166 for saturated and 0.2083 for superheated steam.

TABLE XII

Pipe Sizes for 150,000 Lb of Steam per Hour

Steam	Abs. Press.	Temp.	Specific Vol.	A	Nearest Pipe size in.
Satd.	65	298	6.652	2.771	24
"	115	338	3.878	1.616	18
"	165	366	2.748	1.145	15
"	215	388	2.131	0.8878	14
Suphtd.	315	472	1.604	0.3341	8
"	365	486	1.390	0.2895	8
"	415	498	1.226	0.2554	7
"	465	510	1.098	0.2287	7

A test on a 10-in. line, 1250 ft long, reported in the N.D.H.A. Handbook, p. 230, gives the following:

Btu loss per sq ft per hr for saturated steam—0.226
Btu " " " " " " " superheated " —0.135
Avg. line temperature, saturated steam —400 F
" " " superheated " —450 F

Based on data given in the A.S.H.&V.E. Guide on coverings for pipes, these figures can be corrected approximately for different pipe sizes and different temperatures, as follows:

Increase coefficient $2^1/_2$ per cent for each decrease of 1 in. in pipe diameter below 10 in.
Decrease coefficient 1 per cent for each increase of 1 in. in pipe diameter.

Increase or decrease coefficient 0.1 per cent for each 1 F above or below 400 F for saturated, and 450 F for superheated steam.

Air temperature taken at 50 F.

TABLE XIII

Btu Radiation Loss per Ft of Pipe for 150,000 Lb Steam per Hour

Steam	Abs. Press.	Temp.	Pipe Size	Pipe Sq Ft	Coeff.	Btu Loss per Hr per Ft of Length
Sat.	65	298	24"	6.28	0.171	266.3
"	115	338	18"	4.71	0.194	263.2
"	165	366	15"	3.93	0.207	257.1
"	215	388	14"	3.67	0.214	265.5
Suphtd.	315	472	8"	2.26	0.145	138.3
"	365	486	8"	2.26	0.147	144.8
"	415	498	7"	2.00	0.152	136.1
"	465	510	7"	2.00	0.153	140.8

These tables show that with 400-lb gage steam, as compared with 150-lb gage, the pipe size is reduced from 15 to 7 in. and the radiation loss cut in half.

The percentage radiation loss when transmitting 150,000 lb of steam per hr 1000 ft is 1.43 per cent for 150-lb gage saturated steam, and 0.73 per cent for 400-lb gage steam superheated 50 deg F. With an annual load factor of 30 per cent these losses become 4.76 per cent and 2.44 per cent respectively.

Meters, Temperature Controls, etc.

Present-day meters are more accurate, more reliable and cheaper. Valves, reducing valves, traps and other accessories have been improved and made more rugged. Automatic temperature controls have been developed which operate efficiently and effectively on steam supplied from a distribution system. With purchased steam, when such devices shut off the flow of steam entirely or almost entirely, the customer's expense stops or is greatly reduced whereas, with his own boiler the banking losses still go on, or the boiler operates at a low and inefficient load.

There is a field for very effective cooperation between the steam company and the manufacturer of such devices. Several companies are developing such cooperative action with remarkable reductions in the amount of steam used by individual buildings for heating purposes.

One of the leading steam companies in this development has effected marked savings for its customers by this means, and through active cooperation of their service engineers with the customer. Selling steam at an average rate of approximately $1.10 per 1000 lb, they doubled the amount of steam sold from 1929 to 1932, a remarkable record during the worst of the depression period.

Service of this character not only pays dividends in the form of satisfied customers, but in actual cash return. Consider a company selling 500 million pounds of steam per year, and employing five service engineers at a combined salary expense of $18,000 per year and testing expenses of $20,000. The example cited above indicates that the average rate received for the steam will be from 12¢ to 16¢ per M lb greater than is the usual average when little or no effort is made to render such a service, giving a gross increase in revenue of from $60,000 to $80,000 per year. The possibilities of improving customer relationships and financial returns by selling steam as a service rather than as a commodity, have, in too many cases, been sadly overlooked in the past, and are today just beginning to be generally recognized. Service of this character has long been rendered in the electrical field, and the application of similar methods in the sale of steam should be just as effective as it has been in the electric industry.

Combined Electric and Steam Service

The writer believes that it is in this field that the greatest advantages can be achieved from the developments in high-pressure steam generating equipment and distribution systems. Industrial plants, department stores, hotels, office buildings, to briefly mention the more prominent customers, require electricity for lighting and power, steam for heating, for hot water and for process work, and air conditioning. *High-pressure steam is the only single medium which is available to supply all three of these requirements on the premises.*

Sporadic efforts have been made to accomplish complete service; as for instance, at Rochester, N. Y., Cedar Rapids, Iowa., Wilmington, Del., and in Louisiana. But, in general, two factors have been opposed to developments along these lines; namely, the conviction that the steam business, though often necessary, was nevertheless undesirable, and that all electric current must be generated in large condensing plants.

The previous articles have presented data to show:

1. That steam rate systems are today set up on a basis which is generally fairly commensurate with the value of the service rendered.
2. That the trend today is toward rate systems of the load-factor type, which put the steam companies in a position to take on high load factor business that they previously could not get and thereby to improve their own average load factor.
3. That in this particular field, there is more gross income to be gotten, on the average, from the sale of steam than from the sale of electricity.
4. That present-day high-pressure isolated plants are in a better position than ever before to compete with utility systems which render an electric service only, particularly where there is a use for low temperature heat.
5. That, on the other hand, high-pressure boiler plants and distribution systems enable the utility to render a combined steam and electric service, placing it in a better position than ever before to compete with the isolated plant.
6. That boiler plant and distribution system efficiencies are today markedly higher than the best of the older systems now operating.

The writer believes it is to the advantage of the utility to install as much non-condensing generating equipment as the steam load warrants, up to approximately 15 per cent of the electric peak, and generate as many kilowatt-hours as possible from the steam necessary to meet the steam requirements. It has been shown that in 1929 there were nineteen companies with 95,000-kw generator capacity which generated 168,500,000 kwh. With non-condensing equipment costing $40 per kw against $90 for condensing, and generating a kilowatt-hour on 4000 Btu instead 14,000, this meant a saving of $4,750,000 in investment and a yearly saving in coal of 5810 tons or between $25,000 and $30,000.

A development which has not been touched upon so far, but which is of fundamental importance, is that of the non-condensing turbine. The present-day high-speed units are extremely rugged, very efficient for by-product generation, and can be made practically automatic in their operation. In Rochester, N. Y., units of from 3000 to 5000-kw capacity in both the straight non-condensing and bleeder non-condensing types, equipped with automatic devices to shut them down in case something goes wrong, have been in operation for ten years without a turbine operator in the plant. They are started up by hand and inspected periodically.

In Schenectady, larger units of the mercury type, and non-condensing steam type, are not only operated in a similar manner but are located out-of-doors.

The use of automatic units on high-pressure steam distribution systems, opens up a new field of development in a combined steam and electric service: it offers new opportunities for effective cooperation between the utility and its customer, and new economies and savings for both the customer and the utility.

When rendering a combined service of this character, the effect of the steam demands and the generation of by-product electricity exert a profound influence upon the economics and operation of the electric end of the

business. Following, is a brief discussion of the effect upon:

a. Method of carrying the system electric peak.
b. Method of generating a certain proportion of the kilowatt-hours.
c. Saving in electric system generating plant investment.
d. Saving in electric transmission and distribution system investment.
e. Saving over electric straight condensing plant generating costs.
f. Saving in electric transmission and distribution system losses.

To a utility planning to render such a combined service, two plans of procedure are open:

Plan 1. Boilers operating at 1200 to 1500-lb pressure and high temperature, feeding high-pressure turbines exhausting at 200 or 250-lb pressure into a steam distribution system operating at that pressure.

Plan 2. Boilers operating at 400 to 500-lb pressure with medium superheat, supplying a steam distribution system operating at that pressure, and feeding a considerable number of straight non-condensing or bleeder non-condensing units of from 1000 to 10,000-kw capacity each, equipped for automatic operation and located on the customer's property. In this case periodic inspection and supervision of the units would be provided, and pressure reducing valves installed, with automatic desuperheaters if necessary, to supply steam to the customer in case the turbine should shut down due to failure in the turbine or generator.

In either Plan 1 or Plan 2 as many kilowatt-hours as possible would be generated from all of the steam supplied to the customer, and these kilowatt-hours delivered directly into the main electric distribution system of the utility. In Plan 1, the service pressure at the delivery point to the customer, would be approximately 100-lb gage. In Plan 2 there would be a high-pressure service at 100 lb and a low pressure service at 5 lb.

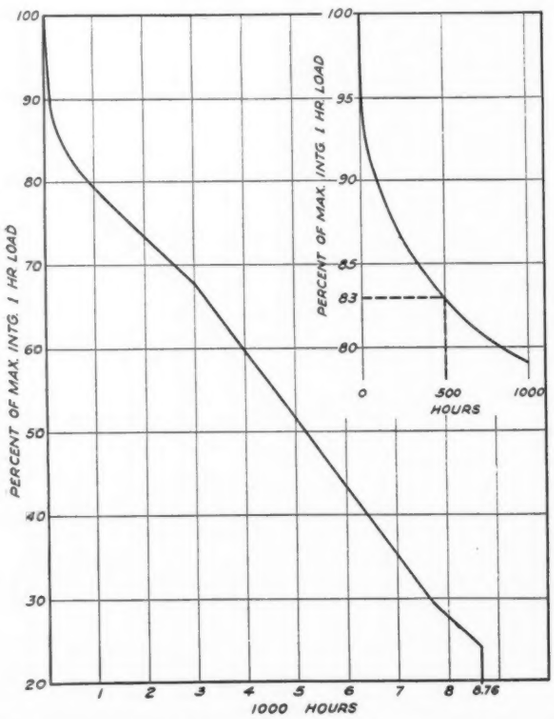


Fig. 3—Typical load-duration curve for an electric system

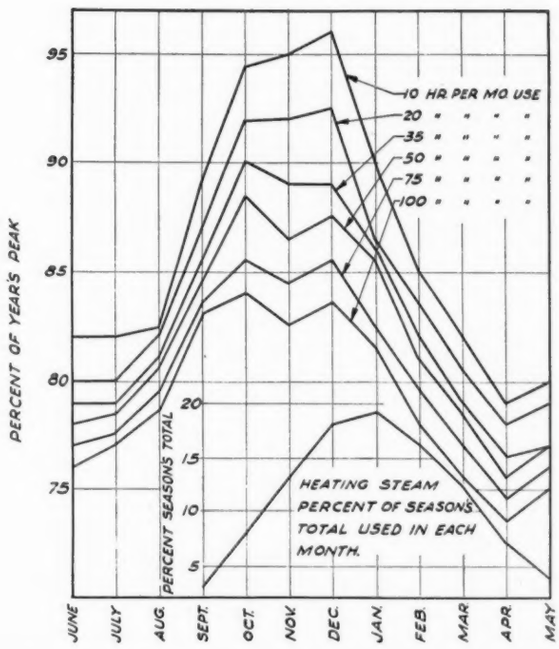


Fig. 4—Curves showing per cent of years electric peak used for specified hours per month for typical electric system

Electric Peak

Fig. 3 shows the load duration curve of a typical electric system, with the upper portion shown to an enlarged scale, and Fig. 4 shows the percentage of the electric peak which is used for 10, 20, 35 and 50 hr per month; also the approximate percentage of the total steam required for space heating used during each of the nine heating months.

A study of these curves shows that during the year, different percentages of the maximum integrated one-hour peak are used for the number of hours per year and during the months of the year shown in Table XIV.

TABLE XIV

% of Int. Peak	Hours Use for Months Shown	Months of the Year
96	10	Dec.
95	20	Nov., Dec.
92	60	Oct., Nov., Dec.
90	85	Oct., Nov., Dec., Jan.
87	200	Sept., Oct., Nov., Dec., Jan., Feb.
85	335	Sept., Oct., Nov., Dec., Jan., Feb.
83	515	Sept., Oct., Nov., Dec., Jan., Feb., Mar.
	The peak used for 10 hr per month is	
89	10	Sept.
94.5	10	Oct.
95	10	Nov.
96	10	Dec.
90	10	Jan.
85	10	Feb.
82	10	Mar.
79	10	Apr.
80	10	May

Note on Fig. 4 that the heating steam curve in general parallels the electric load curves, but is pushed over one month to the right. It is also to be noted that during March, April, May, June, July and August, the ten hours per month use of the peak on the electric system, does not exceed 83 per cent of the maximum winter peak.

These curves show that the maximum heating steam period corresponds, in general, with the maximum elec-

tric load period; therefore, non-condensing electric generating equipment can be installed to meet the demands of the electric peak load period. The kilowatt capacity of such non-condensing equipment will be that amount required to supply the heating and industrial steam up to approximately 17 per cent of the electric system peak.

Such non-condensing capacity will supplant an equivalent of condensing capacity plus any transmission and distribution system electric losses incurred in getting the output of the condensing plant generation to the electric load supplied directly by the non-condensing capacity.

The kilowatt-hours generated by such non-condensing equipment will be produced at a very high efficiency; actual results from such turbine-generators operating at 350-lb gage, show well under 4000 Btu per kwh. The only actual loss incurred is that due to friction and radiation losses in the turbine and generator. The internal losses in the turbine are recovered in the exhaust steam.

The tremendous cost of carrying the top 17 per cent of the duration curve which lasts only about 500 hr out of the year, has been and is one of the serious economic and technical problems of the electric utility. With condensing plant costs running from $80 to $125 per kw, when this investment can be used only from 100 to 500 hr out of the year, means that the fixed charges per kilowatt hour generated will be from 40 to 10 times that of the average kilowatt-hour of the typical electric system.

Combined electric and steam service offers an economical method of carrying this short hour peak, and of generating very cheap kilowatt-hours, thus reducing both the investment required and the operating costs. Non-condensing units of the sizes contemplated can be installed at costs running between one-third to one-half that of condensing plants, and will save from 7000 to 10,000 Btu on each kilowatt-hour generated. They require very little space, can be installed in basements, sub-basements or inexpensive buildings. Modern sound-proofing practically eliminates any objectionable noise. They will be controlled by back-pressure governing devices, automatically taking such electric load as is necessary to supply the steam required by the steam distribution system.

It should not be forgotten that during an emergency, such as the breakdown of a condensing unit, exhausting the steam to the atmosphere is a relatively inexpensive method of operation. Under such conditions these machines would be controlled by the usual speed type of governor with the synchronizing motors all controlled from one point if desired.

The electric transmission and distribution system, like the usual condensing plant, must provide 17 per cent of its capacity for short-hour use. A dozen or fifteen non-condensing units aggregating about 15 per cent of the system peak, located at strategic points in the load area of the distribution system, will form a very reliable substitute for an equal amount of distribution system capacity, resulting in additional savings in investment. The probability of trouble with more than one or two of these units at the same time, if they are given effective inspection and supervision, is very remote.

The kilowatt-hours generated by such machines, in addition to the thermal savings mentioned, will also save transmission and distribution system losses of from 5 to 10 per cent. Better voltage regulation might also be possible.

In conclusion, it has been the purpose of these articles to trace some of the factors in the early development of district steam heating which accounted for the losses sustained and certain prejudices against it, to outline the growth and changes, particularly from about 1920 to 1929, and to point out the improvement in operating results as compared with the earlier experiences; to discuss the improved equipment available today, its effect upon the investment required and operating costs; to point out the relation between steam service and electric service, and the advantages of dual operation; to attempt to picture the present-day possibilities of the development of steam service, not as an unwelcome appendage to the electric utility, but as a co-worker and partner in the task of solving the problems of development and cooperation between the utility and its customer. Even if such a partnership between steam service and electric service is not adopted in most cases, it is still true that a number of steam companies are today making money "on their own," and a steam service designed to meet all the requirements of its customers for steam to be used for power generation, for industrial process work, for space heating and for air conditioning, is well able to stand on its own feet.

Melvin D. Engle has been appointed superintendent of the station engineering department of the Edison Electric Illuminating Company of Boston succeeding George E. Seabury who recently retired.

George L. Knight, vice president in charge of mechanical operations of the Brooklyn Edison Company, has been elected president of United Engineering Trustees, Inc.

Dr. W. F. Durand, past president of the American Society of Mechanical Engineers and Professor Emeritus of Mechanical Engineering, Leland Stanford University, has been awarded the John Fritz Gold Medal for 1936, for notable achievements in hydrodynamic and aerodynamic science, research and education.

Charles F. Scott has been elected chairman of the Engineers' Council for Professional Development, succeeding C. F. Hirshfeld.

Otto de Lorenzi has been appointed Assistant General Sales Manager of Combustion Engineering Company, Inc., as announced by H. S. Colby, General Sales Manager. Except for two years in the American Expeditionary Forces, Mr. de Lorenzi has been with the company since 1916 in various engineering capacities. He is a graduate of Cornell University, a member of the A.S.M.E. and a licensed professional engineer.

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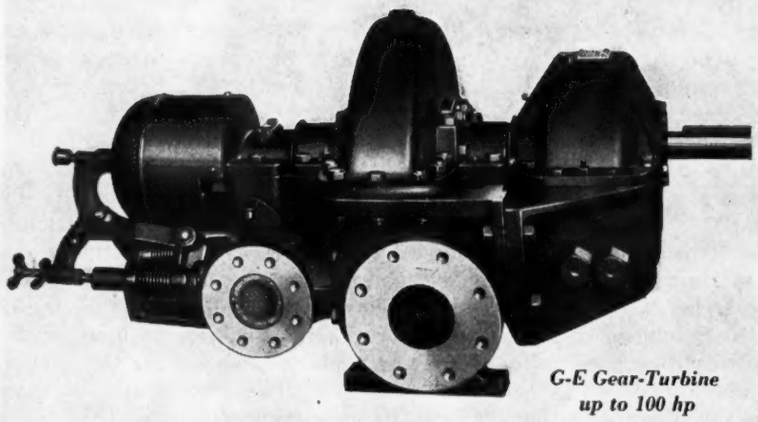

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BOILER DESIGN as Influenced by Factors External to Heat Absorbing Surface*

The term boiler is often entirely inadequate to describe the steam generating equipment. Water walls, water screens, superheaters, economizers and air heaters have been added to increase heat absorbing surface until the boiler finds itself simply an integral part of a steam-generating unit. During this period the boiler designer has been increasingly conscious of the growing importance of this additional heat absorbing surface and of the necessity of combining this variety of equipment into a correctly proportioned unit. These developments have introduced many factors of influence in the trend of modern boiler design, some of which are here discussed.

By OTTO de LORENZI and

L. J. MARSHALL

Combustion Engineering Company, Inc., New York

NOT so very long ago the boiler manufacturer's obligation, in connection with the sale of a boiler, consisted of little more than a guarantee that the boiler had a given amount of heating surface and was designed for a specified pressure. In contrast, the manufacturer is now expected to guarantee such items as capacity, efficiency, exit gas temperature, draft loss, superheat and pressure drops, not only for the boiler, but to coordinate these data as affected by the addition of superheaters, economizers and air heaters. In order to make guarantees of this nature, the manufacturer must be able to predict very closely the performance of each and all of these pieces of equipment. In obtaining information upon which to base his predictions, he has been seriously handicapped through lack of definite operating data. While a thorough knowledge of the theoretical considerations involved is essential, alone it is practically useless. Experience and continued investigation, however, have served to improve the situation.

Probably the most useful means for obtaining information upon which to base performance prediction is offered by reference to the results of carefully conducted boiler tests. It is not practical for a manufacturer to build a boiler and operate it for the purpose of obtaining information upon which to base performance predictions. In the first place the cost would be prohibitive. Secondly, the results obtained would be applicable only to the particular type of boiler tested, whereas the manufacturer desires information to cover a widely diversified field involving many types and sizes, as well as a wide range of conditions. Moreover, it is impossible to approach actual conditions by the use of data obtained on reduced-sized models. However, even in the face of these limitations, it is possible to apply the laws of heat transfer and flow of gases to the results of tests and to obtain thereby data applicable to the entire range of boilers and operating conditions.

Whether in analyzing test results or predicting performance, one of the primary items required is an accurate determination of the weight of gases generated in the furnace and passing over the boiler surface. Obviously, any metering device is impractical for this purpose. The universally adopted method is a determination based upon the analyses of the fuel and products of combustion. Test results show "Orsat" readings indicating the volume percentages of CO_2, CO and O_2 in the flue gases. From the fuel and gas analyses and the use of the so-called "Mol" system, in which the molecular weights of the constituents are used, the quantities of flue gas as well as air required for combustion can be calculated. Another convenient formula in common usage is the following:

Pounds of dry gas per pound of carbon =

$$\frac{11\ CO_2 + 8\ O_2 + 7\ (CO + N_2)}{3\ (CO_2 + CO)}$$

in which the symbols represent the percentages by volume as given by the flue gas analysis. To the dry gas weight must be added SO_2 produced from the combustion of sulphur in the fuel plus the moisture produced from the combustion of hydrogen and the moisture in the fuel itself. Complete derivations of determining the quantity of flue gases can usually be found in any technical book on combustion.

The foregoing explanation is intended to show the method of obtaining the flue gas quantity based upon boiler test results.

* Presented before the Engineers' Society of Western Pennsylvania, Pittsburgh, October 15, 1935.

On the other hand, when making performance predictions, such complete information is not available. The excess air or CO_2 is dependent upon the character of the fuel and the type of combustion equipment used. With these factors known the proper excess air required for good operation can be decided. Although it is not always possible to know beforehand the exact analysis of the fuel, it is highly desirable to know its proximate analysis, calorific value and source. Where the ultimate analysis of the fuel is known, the quantity of air required to burn it can be computed from the formula:

$$\text{Lb of air per lb of fuel} = 11.52C + 34.56\left(H - \frac{O}{8}\right) + 4.32S$$

where C, H, O and S represent the proportions by weight of carbon, hydrogen, oxygen and sulphur.

When the proximate analysis only is known, it is found more convenient to use empirical methods for computing the quantities of air and flue gas.

Fig. 1 illustrates an empirical curve of this type for use with bituminous coals. It will be noted that percentages of excess air are plotted as abscissas with "pounds of gas per million Btu input to furnace" as ordinates. It has been found that this relationship holds with a surprising degree of accuracy for the entire range of bituminous coals. The lower part of the curve provides easy means for obtaining the CO_2 corresponding to any given percentage of excess air. This relationship between CO_2 and excess air is influenced by the ratio of fixed carbon to volatile matter. Actually there should be a family of curves to cover the entire range of fixed ratios of carbon to volatile content, but the spread between them would be so small that the two lines shown—one for high volatile and one for low volatile coals—will give sufficiently accurate results. Curves of this nature can be plotted for virtually all fuels.

Assuming that the gas quantity is known, the next major item required is the temperature of the gas leaving the furnace and entering the boiler.

Many Factors Influence Furnace Temperature and Heat Absorption

Past experience has demonstrated that furnace temperature and furnace heat absorption depend on so many factors that the problem of devising a satisfactory means for their determination is difficult. All of the following factors have some influence on the furnace temperature:

1. Type and characteristics of fuel.
2. Type and rate of firing.
3. Per cent CO_2 or excess air in furnace.
4. Amount and temperature of preheated air.
5. Amount and nature of water-cooled surfaces.
6. Cleanliness of water-cooled surface.
7. Temperature of water and steam within water-cooled surfaces.
8. Design and arrangement of furnace.
9. Losses due to radiation.

With pulverized coal firing and to a lesser degree, oil and gas firing, the entire furnace is more nearly filled with flame than with stoker firing where the flame occupies a limited zone just above the fuel bed. The flame in pulverized coal-fired furnaces contains a large amount of incandescent carbon particles which radiate more heat than does the non-luminous flame of a gas-fired furnace.

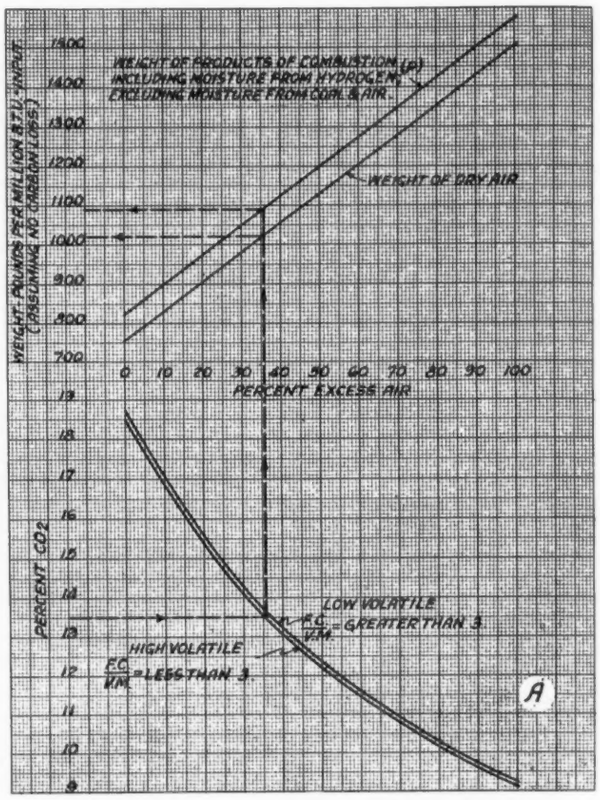
WEIGHT OF PRODUCTS OF COMBUSTION (Lp) INCLUDING MOISTURE FROM HYDROGEN, EXCLUDING MOISTURE FROM COAL & AIR
WEIGHT OF DRY AIR
WEIGHT, POUNDS PER MILLION B.T.U. INPUT (ASSUMING NO CARBON LOSS)
PERCENT EXCESS AIR
PERCENT CO2
LOW VOLATILE F.C./V.M. = GREATER THAN 3
HIGH VOLATILE F.C./V.M. = LESS THAN 3
A

Fig. 1—CO_2, excess air, gas and air weights
(for coal only, except anthracite)

With regard to the other factors, the following comments are in order:

The nature of the water-cooled surface naturally affects its heat absorption. For instance, armored or protected tubes will not absorb heat as readily as bare tubes.

Any comparison of absorption rates should take into consideration the construction of the furnace water walls. For example, plain tubes may be closely spaced, forming an almost continuous metallic wall in which approximately half of the tube absorbs heat by radiation. Plain tubes spaced apart and having the rear half embedded in refractory are also half exposed to radiation. Plain tubes spaced apart, and which either touch the wall or are not away from it, have more than half of their surface exposed to radiant heat from the refractory wall as well as from the furnace. Coatings of ash and slag on parts of the walls cause those parts to be comparatively ineffective.

Temperatures in a furnace operated at constant rating may vary 200 to 300 deg depending on the cleanliness of the water-cooled surface. Accuracy of furnace calculations for slag-covered or dirty furnace walls will depend largely on the judgment and experience of the calculator. In a paper on heat absorption in water-cooled furnaces, presented before the Fuels Division of the A.S.M.E. in December 1930, William L. DeBaufre proposed to take account of heat absorption with clean, slightly dirty and dirty furnace water walls by means of "effectiveness factors" based on test results from various sources.

The design and arrangement of furnace walls also exert an influence on the furnace heat absorption. For example, with stoker-fired furnaces, particularly those hav-

ing chain and traveling grate types equipped with long arches, only a small portion of the boiler surface and upper furnace walls "see" the fire. This results in a reduced furnace wall heat absorption and therefore a high furnace temperature.

With pulverized coal, as well as oil and gas firing, the size of the furnace and the ratio of its plan area to perimeter undoubtedly affect the furnace wall heat absorption. In partially water-cooled furnaces the location of the water wall surface influences the absorption.

Although heat absorption by the furnace walls takes place primarily by radiation as expressed by the Stefan Boltzman law, and is proportional to the difference of the fourth powers of the absolute temperatures of the radiating surfaces, it is evident from the foregoing discussion that there are many other factors which exert an influence. Mr. DeBaufre, in the paper previously mentioned, gave the following reasons why the heat radiated to the furnace walls does not follow the fourth-power law:

The temperature of the burning fuel and products of combustion within the furnace is not uniform, but varies from a maximum at the place of most intense combustion to a minimum generally at the furnace outlet.

The water-cooled surfaces absorbing radiant heat are at a higher temperature than the water evaporating into steam within these surfaces.

Some heat may pass by convection and conduction rather than by radiation.

Radiation from non-luminous gases does not follow the fourth-power law, and radiation from burning fuel may not follow this law.

The flames may not entirely fill the furnace volume.

Clouds of cool, unburned fuel may intervene between the hot flames and part of the water-cooled surface.

Combustion may not be complete within the furnace, but continue into and through the boiler-tube banks.

In attempting to devise a satisfactory means for predicting furnace performance, curves of the form shown in Fig. 2 provide a simple means of determining the furnace heat absorption. This curve represents a plot of test results on a large pulverized coal furnace. It has been found that curves on similar furnaces, when plotted on the same coordinates, are in close agreement. The abscissas represent heat liberated per square foot of projected radiant water-wall surface. The heat liberated is

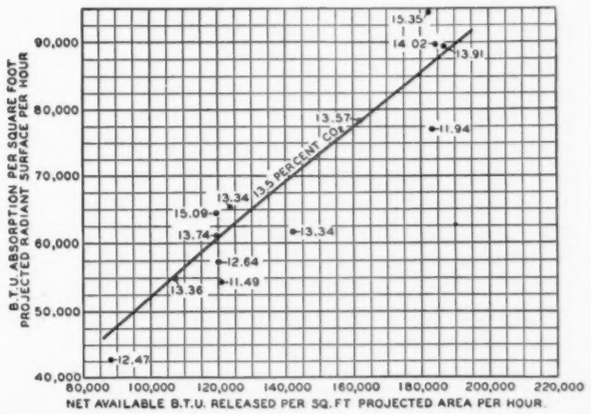

Fig. 2—Furnace heat absorption on large pulverized coal-fired boiler. Figures show per cent CO_2 in furnace

the net heat available in the furnace, which is the sum of the Btu in the fuel and in the preheated air less the furnace losses and less the latent heat of moisture in the fuel and moisture produced by the burning of hydrogen, expressed in Btu. The ordinate of the curve represents the heat absorbed per square foot of projected radiant surface. The numerals at each of the plotted points represent the corresponding CO_2 in the furnace. Curves can be drawn through points of the same CO_2 to represent the absorptions corresponding to liberations for constant CO_2. By means of such curves obtained from the results of tests on a number of different types of furnaces and fuels, it is possible to determine the absorption by the furnace water walls.

Since the difference between the heat released and the heat absorbed in the furnace walls represents the heat content in the gases at the furnace outlet, the gas temperature entering the boiler may be easily determined.

Factors Affecting Boiler Itself

Having considered the determination of gas weights and temperatures entering the boiler, let us investigate the factors which have influenced the design of the boiler itself. Fifteen years ago boilers were rarely operated above 200 per cent rating and difficulty with slag deposit on the first few rows of boiler tubes was therefore not serious. However, the present high rates of combustion, and consequent high furnace temperature and high velocity of gases at the entrance to the boiler, have brought troubles from slag deposits on the boiler tubes, particularly with coals of low fusion temperatures. This trouble has presented more or less serious difficulty on boilers of the straight-tube type equipped with interdeck superheaters and where the entrance to the first pass may be somewhat restricted. By arranging the drop-bank section of the boiler on wide centers both horizontally and vertically, this trouble has been greatly reduced.

Superheating Surface and Its Location

The open arrangement of the first pass of bent-tube boilers is not ordinarily as subject to slagging difficulties, but here also it is often desirable to open up the first few rows of tubes by designing the boiler so that alternate tubes of the first row project into the furnace 8 to 12 in. farther than the normal first row of tubes.

The flue gases, after passing through the slag screen and usually a few rows of boiler tubes, next encounter the superheater. It is only within the past few years that the superheater has received very serious consideration. Experiments have proved steam temperatures of 900 F to be entirely practical. Manufacturers are now offering superheaters for many industrial and utility plants for temperatures ranging from 800 to 950 F. As the temperatures increased, so the required space increased. With the development of alloys to withstand the high temperatures, it has been possible to move the superheater nearer to the furnace, and in this manner, to maintain a higher temperature difference between the steam being superheated and the products of combustion. The result has been to reduce the amount of surface which would have otherwise been required. Nevertheless, a very large amount of surface is required to obtain these high temperatures. The space required to accommodate this surface is so great that it becomes necessary, in many in-

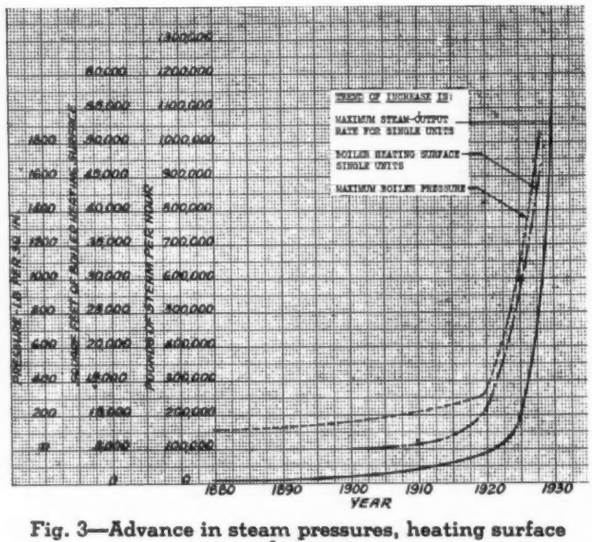

Fig. 3—Advance in steam pressures, heating surface and output

stances, first to design the superheater and then consider the boiler afterward, with the result that it is not unusual to find very little available space remaining for the boiler.

While the development of suitable alloys has been the chief factor in making these high temperatures and pressures a commercial success, the development of suitable means and materials for supporting the superheater has been of almost equal importance. In some cases the elements nearest the furnace have been supported by the elements above by means of lugs welded to the elements themselves and kept sufficiently cool by steam within the tubing.

Means for Maintaining Constant Superheat

A phase of superheater design which has found increased importance of late has been to provide means for maintaining a constant steam temperature over a relatively wide range of the load operation. Several methods of maintaining a constant temperature have been used, all of which affect the design of boiler:

1. The compensating type superheater applied where two boilers are arranged over one furnace, one boiler being equipped with a large convection superheater and some semi-radiant intertube superheating surface while the other boiler has semi-radiant superheating surface only. The temperature is controlled by dampers which regulate the amount of gas passing through each of the two boilers.

2. A second method is by use of de-superheaters to reduce the temperature of the overly-superheated steam to the desired temperature. This equipment is relatively expensive and requires considerable space outside of the boiler setting.

3. A third method which appears to be better adapted to the average boiler is to design the boiler with by-passes and suitable dampers for regulating the quantity of gas passing over the superheater.

In recent years, particularly with large units, the amount of boiler surface placed after the superheater has become smaller and smaller. The use of completely water-cooled furnaces has resulted in a large proportion of total heat absorption by the furnace wall tubes. With steam temperatures elevated to 800 to 900 F, the heat absorbed by the superheater may account for as high as 700 deg drop in the temperature of the gases passing over the surface. The proportion of the total heat which must be absorbed by the boiler is relatively small. However, the mean temperature difference between the flue gas and the boiler heating surface is also relatively small, necessitating the use of a proportionately large amount of boiler surface, in order to reduce the exit gas temperature to a reasonable figure. Particularly is this true at the higher pressures where the temperature of saturated steam is high. In such cases the use of larger economizers is found to be more economical.

The boiler absorbs heat primarily by convection. While much research and experimenting has been done to determine the rates of heat transfer applicable to boilers, these data assume ideal conditions, which are never encountered in actual operation. Virtually all of the experimental results indicate that the convection transfer rate is a function of the mass velocity of the gas, and most of the proposed formulas include terms for adjusting the transfer rate for varying temperature conditions.

Deviations from Formula Necessary

In attempting to predict the exit gas temperature from the boiler it is necessary to allow for deviations from the proposed formulas for the following reasons:

1. The mass velocity is not uniform.
2. Temperature of gas is not uniform.
3. Some radiant heat from the furnace finds its way into the boiler passes.
4. Some radiation takes place from the non-luminous CO_2 and water vapor even at the relatively low temperatures encountered in the boiler.
5. The entire tube heating surface is not effective. With any practical baffling arrangement some of the surface is by-passed.
6. The gas flow is not all transverse to the tubes in the case of straight-tube boilers or all parallel to the tubes in the case of longitudinally baffled bent-tube boilers.

Investigation of test results on existing installations indicates that heat recovered by the surface near the boiler outlet is a relatively small amount. Studies have been made with various baffling arrangements in an attempt to arrange the surface and baffles to obtain a maximum transfer of heat with bent-tube boilers by use of cross baffles. In many cases it has been found possible to obtain considerably better performance by such an arrangement of baffles.

During the past several years elaborate studies have been made to determine the most economical design of steam generating units. This does not mean most economical from the standpoint of first cost, but rather with a view of securing the greatest return on the investment, giving due consideration to the expected operating conditions and all other related factors. Modern boiler units are the result of exhaustive studies of all of these factors.

At this point it may be of interest to consider briefly the advancement in increased boiler pressures, heating surface and steam output as illustrated by Fig. 3. It is evident that progress in this direction was negligible from 1880 until 1920, at which time, coincident with the end of the World War period, occurred the advent of the

commercial application of pulverized fuel firing to steam boilers. The water screen, the water-cooled wall and the air heater soon entered the picture and the period of rapid advancement in steam generation as we now know it was on its way.

Before consideration can be given to the design of the boiler units themselves, the governing conditions must be ascertained. These include: the normal and maximum outputs required; the character of the daily or periodic load, the yearly load factor and the character and cost of fuel. In addition, two other factors are becoming increasingly important—availability and reliability. Not many years ago it was customary to provide for boiler outages by the addition of one or more spare units to carry the load when some of the boilers were shut down for cleaning, repairing or rebuilding the furnaces, which incidentally, were not infrequent occurrences. But with the completely water-cooled furnaces now extensively used and with the present knowledge and adoption of proper methods of feedwater treatment, outages from these causes are rare indeed. Many of the newer installations are operating with availability factors well over 90 per cent. Such records are possible only by virtue of proper analysis of the problem in hand and due consideration to all the factors influencing the design of the unit.

Some Economic Considerations

Where reliability and availability are important considerations, it may be necessary to increase the initial capital investment sufficiently to allow the incorporation of certain features which will assure their accomplishment. For example—to provide a boiler with a suitable slag screen and a large furnace, may add to the cost of the unit, but merits ample justification in the definite elimination of a potential source of periodic shutdowns. The size of the furnace and extent of the slag screen will largely depend upon the characteristics of the coal and the method of firing. A large furnace reduces combustion rates, provides additional space for flame propagation and reduces the temperature of the products of combustion entering the boiler tubes. The slag screen provides additional cooling and reduces gas velocities, thereby causing the semi-molten ash particles carried in suspension to solidify in this zone before entering the main tube bank of the boiler.

One of the next steps in the design is the determination of the extent to which high efficiency may be sought with suitable return on the investment. Naturally this cannot be estimated until the increment cost of obtaining additional efficiency is known. It is here that the boiler designer is confronted with the problem of proportioning the various heat-absorbing surfaces to show the highest efficiency for the least cost. Needless to say, the method of arriving at the answer is one of "cut and try." The efficiency which can be justified will depend upon the cost of obtaining it and this cannot be decided until the amount and cost of the various surfaces have been determined. The cost of a particular combination of surfaces must be evaluated and, if found not justified, other combinations must be similarly considered until the proper one is attained.

The proper combination of surfaces depends principally upon these items:

1. Pressure.
2. Feedwater temperature.
3. Type of fuel.
4. Type of boiler.

In some instances the temperature of preheated air is limited by the firing equipment and it is then only necessary to determine the economical combination of water walls, boiler and economizer surfaces.

The cost of boiler surface is largely dependent upon the type of boiler used. It is usually possible to increase the surface of bent-tube boilers by increasing the vertical drum centers. If this can be done without increasing drum size, the cost of adding surface is comparatively small. Adding surface to straight-tube boilers usually means additional length of headers which naturally are more costly than tube surface alone.

The amount of heat transferred depends upon the temperature difference between the gas and water. This temperature difference as applied to the surface at the boiler outlet depends upon the saturated steam temperature within the boiler. In the economizer which is arranged for counterflow of gas and water the temperature difference applying is that between the gas and feedwater being heated. Economizer surface can therefore be substituted for boiler surface and recover more heat because of the increased temperature difference, which condition applies until the feedwater is heated to the temperature of saturated steam. For high pressures and comparatively low feed temperatures it is possible to justify large amounts of economizer surface. The economizer also can be arranged for any desired gas velocity and, with practically all of the surface effective, whereas the boiler surface, because of its location and the multiplicity of passes which must be employed, cannot be made 100 per cent effective. Furthermore, it is often not feasible to arrange the boiler baffles to give the desired gas velocity. The economizer, on the other hand, may be arranged for staggered flow transverse to the tubes, which gives a very effective transfer of heat to the water.

The increased use of economizer surface in recent years serves to indicate that a thorough analysis has repeatedly justified more economizer surface and less boiler surface as an economic expedient. In addition, the space required for the economizer is less than for an equivalent amount of boiler surface which is often an important consideration.

The steel tube economizer is almost universally used at present. Its use has been brought about by the extensive employment of high-pressure steam and the development of manufacturing methods for making short radius return bends as well as forged return bends.

In conclusion, it may well be pointed out that a most important feature of the advance during this period is the knowledge and experience which have been gained. As a result, the boiler designer proceeds with greater confidence in his efforts to combine the several related elements of steam-generating and heat-absorbing equipment into a properly proportioned unit, and with increased assurance of greater certainty of ultimate results.

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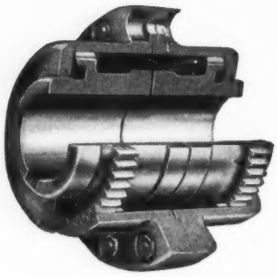

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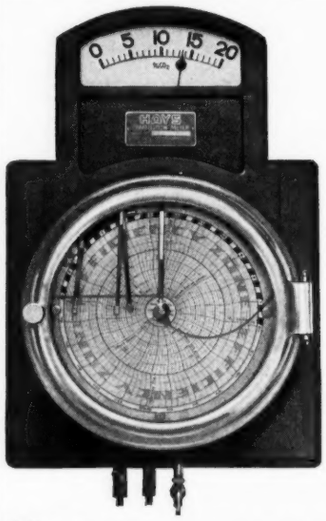

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Factors in Selection of Coal for Underfeed Stokers*

The purpose of this paper which will be presented in full at the Fuels Session of the A.S.M.E. Annual Meeting, Dec. 2–6, 1935, is to point out the complexity of the problem of selecting coals for underfeed stokers. It does not attempt to present a method or a formula for such selection, but aims to stimulate discussion.

AN analysis of data on 7284 industrial plants, accumulated in a market survey recently conducted by Appalachian Coals, Incorporated, reveals the prominent position of the underfeed stoker. This survey covered the principal market areas for southern high volatile coals and included approximately 15,000 coal-burning units in industrial plants located in Georgia, eastern Indiana, central and eastern Kentucky, lower peninsula of Michigan, North Carolina, that part of Ohio west of the Sandusky-Galion line, South Carolina and western Virginia. Table I is a summary of the coal-firing equipment in use in the plants covered by the survey.

Certain outstanding facts are immediately apparent upon inspection of Table I. The total coal involved annually is 19,634,266 tons; of that amount, 9,429,375 tons or 48.1 per cent is burned on underfeed stokers. Out of a total of 2,532,692 boiler horsepower, underfeed stokers account for 1,139,292 boiler horsepower or 45 per cent. The number of underfeed units is 4423.

The average horsepower of the 14,532 boilers is 174.3, whereas boilers fired with underfeed stokers average 257.6 horsepower.

Of special interest is the large number of hand-fired boilers, numbering 8878 units, with an average of 106.8 horsepower. From a coal-consumption standpoint, these hand-fired boilers rank next to underfeed stokers with an annual consumption of 4,089,827 tons, or 20.8 per cent of the total. Together the underfeed stokers and hand-fired grates account for 13,519,202 tons, or approximately 70 per cent of the total coal involved in the survey.

This survey reveals that the average horsepower of the underfeed stoker-fired units is rather low but that these units are very numerous. This probably indicates the effect of the great increase in small industrial underfeed stoker installations made in recent years. The data on underfeed stokers by types (Table I) show that from a tonnage standpoint, the multiple-retort stoker accounts for 6,587,655 tons or 69.9 per cent of the total coal burned by underfeed stokers; the average boiler horsepower per unit is 491. The ram-feed stoker accounts for 1,997,690 tons of 21.1 per cent with an average boiler horsepower of 220. Together, the multiple-retort and the ram-feed underfeed stoker account for 8,585,345 tons or 91 per cent. The screw-feed underfeed stoker accounts for 844,030 tons or 9 per cent of the total coal burned on underfeed stokers.

Approximately 80 per cent of the coal shown in the survey is mechanically fired and the balance, 20 per cent, is hand-fired. Coal mechanically fired is proportioned on a tonnage basis approximately as follows: Underfeed stokers, 60 per cent; pulverizers, 20 per cent; chain-grate stokers, 11 per cent; overfeed stokers, 7 per cent; and spreader stokers, 2 per cent. The multiple-retort stoker alone accounts for approximately 40 per cent of all the coal mechanically fired in the plants covered by the survey.

Present Basis for Coal Selection

The selection of coals is not, as a rule, given adequate consideration. The most common, and undoubtedly the poorest, method is that wherein the buyer, frequently disregarding the origin of the coal, bases his selection on the proximate analysis and calorific value. Even though the coal may rank high in heat units and low in ash, moisture, sulphur, etc., and the buyer may be well pleased with the negotiations, it is only by chance that he has made a good selection. This is simply blind buying of heat units over the counter, and it does not make for ultimate satisfaction with respect to performance and economies. The proximate analysis and calorific value give important information and are useful in checking shipments of a coal which has been found to give satisfaction, but by themselves they are inadequate indices of the ability of a coal to meet given requirements.

The trial-and-error method of selection, in which the power plant engineer acts as judge, basing his conclusions upon results obtained, is a much more reliable and satisfactory basis for selection. In fact, there is nothing seriously wrong with the trial-and-error method except that it is sometimes an expensive and long-drawn-out procedure. In the end, it will usually be productive of good results, and coals which meet the plant's specific requirements will be obtained.

The average coal buyer does not appreciate the complexity of the problem of coal selection, but is rather inclined to rely upon some certain technique which he has developed, couched in good intentions, but likely based

* Submitted by a committee of Fuel Engineers representing Appalachian Coals Inc., consisting of R. L. Rowan, Fuel Engineer, General Coal Company, Philadelphia; L. A. Shipman, Combustion Engineer, Southern Coal and Coke Co., Knoxville, Tenn.; T. H. Queer, Sales Engineer, Pittsburgh Coal Co., Pittsburgh, Pa.; F. J. Kasper, Combustion Engineer, The Koppers Coal Company, Philadelphia, Pa.; O. O. Mallers, Inspection Division, Appalachian Coals, Inc., Cincinnati, O.; J. E. Tobey, Manager of Fuel Engineering Division, Appalachian Coals, Inc., Cincinnati, O.

upon erroneous and unsound principles. When this technique fails to produce satisfactory results, it creates an additional expense in the operation of his plant.

If the error in selection is not too great, the plant may operate for a period of years without this error becoming apparent. On the other hand, if it shows up in high costs, the buyer, due to lack of understanding, may place the blame entirely upon the coal or the equipment instead of upon the inadequacy of the system of selection. In many such cases, the coal may have been exactly as represented by the seller, but it was called upon to meet a set of conditions for which it was unsuited; or, perhaps, its poor performance was due to lack of knowledge on the part both of the buyer and the seller as to the proper firing technique. There appears to be a gap, or "no-man's-land," between the knowledge of coal application possessed by buyer and seller which adversely affects their common problems.

The principles involved in underfeed combustion are not generally well understood. Superior coals improperly fired may show inferior performance to poor coals properly fired. There is a best technique for burning each coal and that technique varies with each type of equipment and with different load conditions. As a rule, coals which give the least trouble in given stoker installations usually prove to be the most economical coals to use, and, conversely, coals which cause serious trouble and result in high maintenance are usually neither efficient nor economical to use. Excessive maintenance usually reflects poor combustion and undesirable conditions, not the least of which is a poor reliability factor and frequent boiler outage.

In the present state of the art of fuel burning, no set of specific rules can be compiled which will be flexible enough to cover the many situations involved in the selection of coals.

Furthermore, the art of fuel burning is undergoing constant change. Coal preparation, equipment design, plant operation, and other kindred factors are still far from being standardized.

Preparation of Stoker Coal

Preparation of coal by sizing and cleaning, frequently alters its burning characteristics. This is important and should command the interest of those engaged in the design and operation of plants consuming coal, as well as those having charge of the purchase of coal.

Since this paper deals with the selection of coals for underfeed stokers, the discussion will be limited to those sizes of coal which are now generally sold for this purpose. In the coal trade, these sizes have usually been known as slack, nut and slack, or nut, pea and slack coal. The terms have often been used interchangeably but in this discussion the term "nut-slack" will generally be used. Formerly run-of-mine coal was the predominating size purchased but this size had to be crushed before going to the stoker.

The nut-slack size consists of the smaller pieces of coal produced in mining, handling and preparing of coal for market. Round-hole or equivalent openings are used in the screens and the diameter may range from the equivalent of a $^3/_8$-in. round hole up to as large as $2^1/_2$in., $2^3/_4$ in., and, in a few cases, to 3 in. Usually, however, the screen openings do not exceed $2^1/_2$in. round, while the prevailing sizes of screen openings seldom exceed 2in. round. The nut-slack size consists of all coal passing through the screen, and, therefore, contains a range of sizes from pieces which will just pass through the screen opening down to nearly zero in size. A large number of nut-slack coal sizes are produced throughout the United States, probably far too many different sizes.

In many mines, due to favorable seam and mining conditions, it is possible to produce the low-ash products even in the smaller sizes by the usual mining and screening operations. In some mines, however, it is not practical, in the usual mining and screening operations, to produce products sufficiently low in ash or sulphur or both to meet certain market requirements. Whether the quality of such coal can be sufficiently improved by mechanical cleaning will depend largely on whether the impurities are free or can be freed from the coal and also whether the impurities have a sufficiently high specific gravity. Every mine requires a special investigation and study.

The cleaning of coal is a complex subject but the quantity being cleaned is steadily increasing as shown by L. N. Plein's report on "Mechanical Cleaning of Coal 1929 to 1934" (*4*)[1] which says:

"Despite a period of depression the mechanical cleaning of bituminous coal has shown a steady and persistent upward trend as indicated by the increase in the percentage mechanically cleaned from 6.9 per cent in 1929 to 11.1 per cent in 1934. The tons of cleaned coal dropped from a high point of nearly 39,000,000 tons in 1930 to a low point of 30,000,000 tons in 1932, coincident with the

[1] Numbers in parentheses refer to bibliography at end of paper.

TABLE I
SUMMARY OF COAL FIRING EQUIPMENT†

Firing Equipment	Number of Units	Installed Boiler Horse Power	Boiler Horse Power Per Unit	Total Annual Coal—Net Tons	Per Cent of Total Coal Used	Coal per Year, Net Tons: Per Boiler H.P.	Coal per Year, Net Tons: Per Unit
Hand-fired grates	8,878	949,220	106.8	4,089,827	20.8	4.3	461
Underfeed stokers	4,423	1,139,292	257.6	9,429,375	48.1	8.3	2,130
Overfeed stokers	517	124,130	240.1	911,130	4.6	7.3	1,762
Chain grate stokers**	371	151,815	409.2	1,749,010	8.9	11.5	4,714
Pulverizers	240	144,730	603.0	3,212,124	16.4	22.2	13,384
Spreader stokers	103	23,505	228.2	242,800	1.2	10.3	2,358
Total or average	14,532	2,532,692	174.3	19,634,266	100.0	7.8	1,351

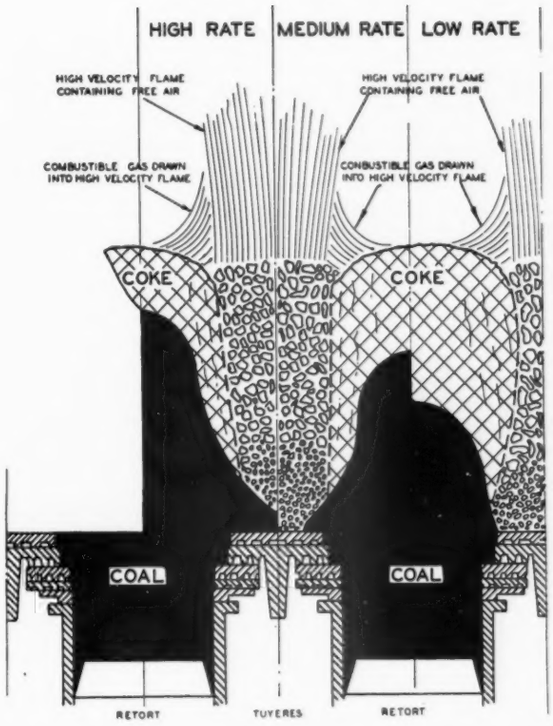
** Includes all traveling grate stokers.

DATA ON UNDERFEED STOKERS BY TYPES

Screw feed	1,605	136,390	85.0	844,030	9.0	6.2	526
Ram feed	1,408	309,742	220.0	1,997,690	21.1	6.5	1,420
Multiple retort	1,410	693,160	491.0	6,587,655	69.9	9.5	4,675
Total or Average	4,423	1,139,292	257.6	9,429,375	100.0	8.3	2,130

† Taken from report "Industrial Coal-Burning Equipment," by Appalachian Coals, Incorporated, Cincinnati, Ohio, 1935.

NOTE: A number of plants with hand-fired grates or underfeed stokers installed in low pressure heating boilers having no horsepower rating are included in this summary. These plants are counted in under the column "Number of Units." The figures shown for boiler horsepower per unit and coal per year per boiler horsepower opposite the classifications for "Hand-fired Grates" and "Underfeed Stokers," and in the totals, would be modified, of course, had these plants been omitted.



Underfeed burning characteristics with different combustion rates

Cross-section taken at upper end of underfeed stoker
Dark areas—coal
Cross-hatched areas—coke in process of formation
Separate particles—coke undergoing rapid combustion

low production of bituminous coal in that year. Since 1932, the growth has become rapid again. Preliminary figures for 1934 indicate that the tonnages cleaned that year will exceed the boom year of 1929 by at least 2,900,000 tons."

What Is Thought to Occur in an Underfeed Stoker Fuel Bed

A concerted effort is being made at the present time to fathom and explain the behavior and peculiarities of coals burned on underfeed stokers. The work done by the Bureau of Mines, as given in its Bulletin 378, has established the fundamental principles of the actions that occur in underfeed burning, and has shown that the action of igniting the fuel plays a much more important part than it does in an overfeed bed. The report emphasizes that actions in underfeed stokers must be much more complicated than those observed in the test furnaces which were of the pure underfeed type because the motion of the fuel must be very complex and will be controlled by a number of factors which will vary with the design of the stoker and the properties of the coal. As the length of the stoker increases, the distribution of the fuel over the bed becomes better defined; as the green coal passes down the retorts, it must be spread—partially at least—over the tuyères.

In a communication, P. Nicholls, of the Bureau of Mines, suggests that the distribution of fuel on an underfeed stoker at a cross-section may be somewhat as is shown in principle by the illustration, which includes views of high, medium and low rates of burning. The figure suggests that the quantity of green coal which will be over the tuyères will increase with the rate of burning and that at the same distance down the stoker the thickness of the cap of coke over the retort will decrease with increase of rating. It would also be expected that the thickness of the cap at a given rate of burning will increase with the distance from the front wall.

Such a conception would account in a great measure for the blowing of fuel beds and high carbon losses through the boiler at high burning rates as fine uncoked coal is more susceptible to this action than is coke. Above the fuel bed there is illustrated the possible mixing of the distilled gases with the air for combustion. As the coal is pushed into the retort, the top layer ignites, distillation of volatile matter starts, coal cokes and progressively burns. The rapidity of ignition will vary with the individual coal and plays an important part in its performance.

The coal must coke and the coke must break up over the tuyère rows in order to maintain active burning lanes above the tuyère rows. The breaking up of the coke formed from the coal on a multiple-retort stoker is not affected to any great extent by the mechanical functioning of the stoker. The movement of the fuel by the main, or feeding, or secondary rams serves to distribute the coal, but produces very little mechanical agitating effect on the burning fuel. The breaking up of the fuel is dependent on the reactions of the fuel itself. Coals which swell considerably upon coking break up from internal stresses set up by this swelling. Coals which swell only slightly do not break up to such an extent.

Some coals form hard, dense coke masses that fracture or break up much less readily than others, and as a result offer considerably more resistance to air flow. Some form small friable masses that are easily blown off the stoker, whereas others contract after initial coking and swell, thereby causing fissures through the coke masses. Therefore, it will be seen that the condition of the coke over the tuyère rows will depend largely on the character of the fuel itself and is a primary factor affecting the resistance or porosity of the fuel bed.

If the physical and chemical reactions within the fuel bed produce coke in large masses, there will naturally be large interstices between these masses. The resistance to air flow will be low with a thin fuel bed and it may be necessary to carry a thick fuel bed in order to obtain sufficient contact of the air with the coke to maintain the desired proportion of excess air. If the coke is fragile and is broken down into comparatively small pieces, there will be comparatively small interstices between the pieces of coke and the resistance of the fuel bed will be high, and it may be necessary to carry a thin fuel bed in order to get a sufficient amount of excess air through the fire. For example, consider two extremes—Rocky Mountain sub-bituminous and Pocahontas coals.

With the sub-bituminous coals, there is no coking and the fuel spreads over the tuyères practically as green coal. The volatile matter is very readily released, leaving small particles of carbon behind. With such a coal, it is necessary to carry a fuel bed only four to six inches thick over the tuyères. Thicker fuel beds simply result in matting of the fuel and resultant non-uniformity of resistance or operation with a deficiency of air, with consequent excessive fuel bed and furnace temperatures and losses due to incomplete combustion.

On the other hand, when burning Pocahontas coals, particularly on short stokers and at low rates, large masses of coke are formed. It is necessary to maintain a fuel bed upward of two feet thick over the tuyères in order to obtain the desired excess air.

The coke deposited on the tuyères at the lower end of the underfeed section may be of a different character than that of the upper end of the stoker. Frequently, the coke over the tuyères at the upper end of the stoker will be small in size and that at the lower end will be large. Under such conditions, it is necessary to carry a considerably thicker fire at the lower end than at the upper end.

The rate of combustion has a pronounced effect on the character of the coke produced during the operation of the stoker. Usually the pieces of coke will be larger and the fuel bed thicker, at low rates of combustion. At higher rates, where the coke is formed more quickly, the pieces may be very small and it may be necessary to carry a very thin fuel bed. It is usually necessary to set up widely different methods of operation to obtain satisfactory performance over any great range of combustion rates.

At one stage during the coking process the coal is in a plastic state and it is now believed that the plasticity or plastic range (the temperature range through which coal is in a plastic state) has much to do with the performance. It is apparent that coals having a long plastic range will offer a greater resistance to air flow than those having a shorter range.

One common complaint in connection with multiple-retort stoker operation, particularly the older types, is fuel-bed matting; that is, matting of the fuel from retort to retort across one or more tuyère rows. Such a condition is the result of some interference with uniform air flow through the fuel bed which in turn causes the fuel to cake and coke without burning. In its initial stage, this mat may consist of green coal but as it moves downward it is transformed into a dense coke mass. This matting action has a paralyzing effect upon the upper and mid-section of the stoker, throwing a greater burden on the lower end, thereby greatly disturbing its function. The maintenance of a thinner fuel bed tends to prevent this occurrence.

Many times, engineers are confronted with the bewildering fact that two different coals from the same seam, analyzing approximately the same, produce entirely different results both from an operating and efficiency standpoint. Table II shows results secured in a multiple-retort stoker-fired plant under similar load conditions on two groups of coals from the same seams (with the exception of mines 4 and 4A) having similar proximate analyses. Coals in Group 1 produced satisfactory results whereas those in Group 2 caused trouble from fuel bed matting or clinkering. Group 2 coals give high resistance to air flow. Although the cause of this high resistance was not determined, it probably was due to the size of the coal or to the nature of the coking process, possibly involving the plastic range and the manner in which the volatiles were released. The ash-softening temperature of all coals involved was relatively high, and the fact should be noted that despite this, clinker trouble developed with two of the coals. This no doubt was caused by "hot spots" or torch action as a result of the matting of the fuel.

Ash Softening Temperature and Clinker Formation

The ash softening temperature is usually the most important characteristic of ash. Much work has been done on this subject, particularly by the U. S. Bureau of Mines in developing the present A.S.T.M. standard method (*7*) and in determining ash softening temperature for large numbers of American coals. These results are available in various publications (*8*) of analysis data.

Later, work was done by both the U. S. Bureau of Mines (*9*) and the Carnegie Institute of Technology (*10*) on the relation of the present ash softening temperature methods to clinker formation. One of the conclusions of these investigations was "that no other single value for fusibility of the ash would give better coordination with the clinkering scale than does the softening temperature." As a result of recent studies (*11*, *12*), it has been shown that the ash "softening temperature, although empirical, has been found to be definitely related to ash composition." (*13*)

There is some difference in ash softening value as affected by coals with different burning characteristics. For instance, the minimum ash softening temperature of a free burning coal can be slightly lower than that of a strong coking coal in underfeed stoker operation. This is probably due to a freer air distribution with the former coal and to localized heat or zones of higher temperature produced in the fuel bed by the strong coking action of the latter. Underfeed stokers are rather sensitive to the ash softening value and it is necessary to consider this as a factor when designing boiler units. The newer types of underfeed stokers with continuous ash discharge are not as susceptible in this respect as are the older types.

Whether a single, generally accepted size of nut-slack coal for underfeed burning is possible for all kinds of coal and all types of underfeed stokers is, of course, very

TABLE II

SHOWING TWO GROUPS OF SIMILAR COALS WHICH GAVE DISSIMILAR RESULTS ON THE SAME UNDERFEED STOKERS

Mine No.	Evaporation	Clinker Formation	Burning Nature	Casting Loss	As Received, % Moist.	Dry Basis, % V.M.	% F.C.	% Ash	% Sul.	Btu per Lb	Ash Soft. Temp. °F	Explanation
GROUP 1												
1	Good	None	Coking	Normal	3.9	33.3	59.7	7.0	0.7	14270	2750	Satisfactory
2	Fair	None	Coking	Normal	3.0	32.4	61.8	5.8	0.9	14520	2900	Satisfactory
3	Good	Some non-adhering	Coking	Normal	3.9	34.7	57.0	8.3	0.6	13950	2840	Satisfactory
4	Good	None	Coking	Normal	...	35.9	58.3	5.8	0.5	14100	2950	Satisfactory
5	Good	None	Free	Normal	3.7	36.7	57.0	6.3	0.7	14010	2450	Satisfactory
6	Good	None	Free	Normal	8.4	35.1	54.6	10.3	0.4	13340	2900	Satisfactory
	Average analysis				4.6	34.7	58.0	7.3	0.6	14030	2800	
GROUP 2												
1A	Fair	None	Hard coking	Normal	...	35.5	58.0	6.5	0.6	14200	2750	Fires matted
2A	Fair	None	Hard coking	Above normal	...	33.1	56.7	10.2	0.5	13860	2930	Fires matted
3A	Poor	Some-adhering	Hard coking	Excessive	3.4	37.6	55.5	6.9	0.7	14140	2900	Fires matted—clinker trouble
4A	Poor	Some	Hard coking	Above normal	6.4	37.0	55.1	7.9	0.6	13820	2900	Fires matted
5A	Fair	Bad	Caked	Excessive	5.4	37.4	54.0	8.6	0.9	13670	2620	Fires matted—clinker trouble
6A	Poor	Bad	Caked	Excessive	4.0	36.8	53.2	10.0	0.8	13460	2700	Clinker trouble
	Average analysis				4.8	36.2	55.4	8.4	0.7	13860	2800	

Mine numbers 1–1A, 2–2A, et cetera, indicate similar coals from mines in the same seam with the exception of 4–4A which are similar coals from different seams.

doubtful. Also the determination of any such optimum size or sizes would require burning tests on commercial stokers under regular plant conditions which would usually be difficult to arrange. However, despite the difficulties, attention is being given the question of proper size of coal for underfeed burning but so far not enough data are available to draw definite conclusions.

A serious mistake has frequently been made, especially in the industrial field, in not taking into account the fundamental principle that low-grade coals require the highest grade combustion equipment. In contemplation of the use of low-grade coals, design must not temporize with equipment cost. It must include the higher investment necessary to secure proper furnace and stoker design and arrangement of same. High-grade coals having greater flexibility do not necessarily require as large a capital investment in equipment. Between these extremes in coals, design can be modified according to the types of coals involved.

What are the variations in the plant design that result in such differences in results? The variations are found in the following factors:

1. Furnace design—including width, height, depth of furnace and arrangement of furnace volume, whether refractory or water cooled walls, or a combination of the two, type of refractory problem, if any, as a result of design (slagging, spalling or accumulative).
2. Boiler design—shape and size affecting furnace and stoker dimensions, tube spacing, arrangement of gas baffles, draft loss and type, if any, of slag screen.
3. Stoker design—type of tuyères, pusher block or ram movement, as to sequence and extent (whether in phase or out of phase movement), type of agitation and effect on fuel bed, type of ash disposal equipment (single or double dumps, clinker grinder or other continuous ash discharge), method of combustion air distribution.
4. Miscellaneous factors—available stack draft, available forced draft, draft restriction, if any, combustion air temperature (whether normal or preheated), furnace temperatures (particularly at lower end of stoker), coal burning rates, air control (manual or automatic), whether secondary air has independent control, furnace heat release, grade or quality of plant maintenance.

Proof of the sensitiveness of coal and the importance of careful design lies in the fact that the combustion behavior of a coal many times differs in plants of similar design.

The logical explanation for this situation is that other factors, many times considered insignificant, combine and exert a powerful influence upon the behavior of given coals. Slight differences in load conditions, firing technique, air pressure, character of maintenance, etc., appear to create changes in the results given by many coals out of proportion to the magnitude of the change in conditions.

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(11) Demorest, D. J., and Schaefer, R. L., paper presented before Gas and Fuel Division, American Chemical Society, New York, April 22–26, 1935.
(12) Thiessen, G., Ball, C. G., and Grotts, P., Paper presented before Gas and Fuel Division, American Chemical Society, New York, April 22–26, 1935.
(13) Carnegie Institute of Technology, Bulletin 62.
(14) *Trans.* A.I.M.E., Vol. 74, 600 (1926).
(15) U. S. Bureau of Mines Bulletin 344.
(16) *Ind. Eng. Chem., Anal. Ed.*, Vol. 3, 43 (1931).
(17) A.S.T.M. *Proceedings* Vol. 34, Part I, p. 457 (1934). Reprint A.S.T.M. Standards on Coal and Coke 1934, p. 58.
(18) *Power Plant Engineering*, March 1934.
(19) *Power Plant Engineering*, March 1935.

World Power Conference Next September in Washington

As previously mentioned in these columns President Roosevelt extended an invitation last summer to hold the Third World Conference in this country next Fall. This invitation has now been accepted by the International Executive Committee and the date has been set for September 7 to 12 at Washington, D. C. This is not to be confused with the Chemical Engineering Congress of the World Power Conference which is scheduled for London next June.

It is understood that the theme of the Washington meeting will be "national power economy" and that regional planning, fuel conservation, water power and regulation of public utilities will be among the subjects to be discussed.

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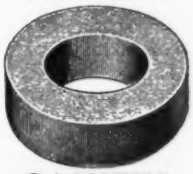

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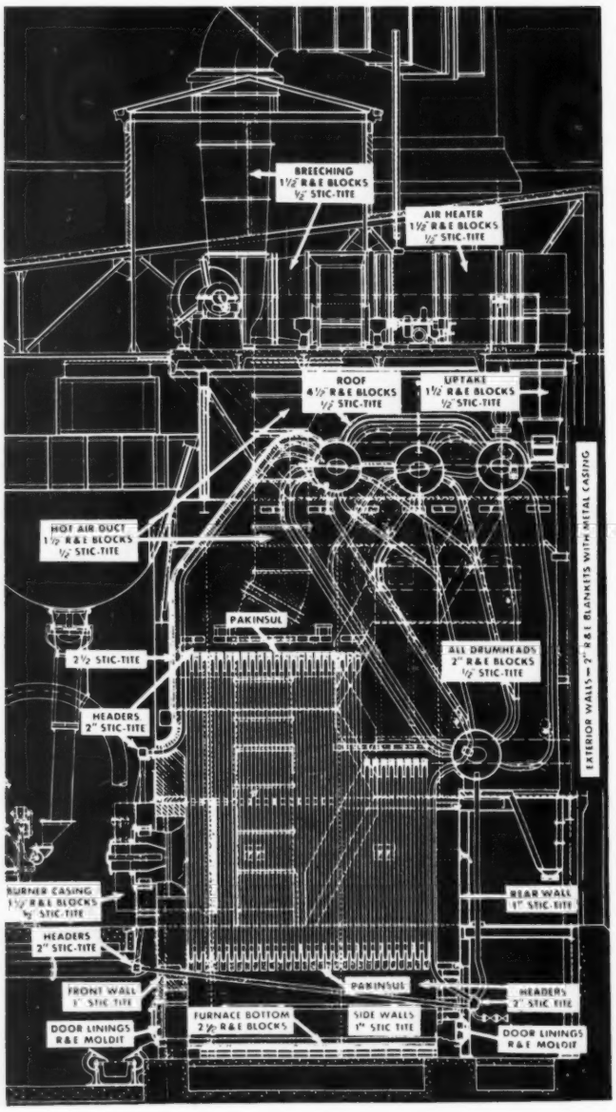

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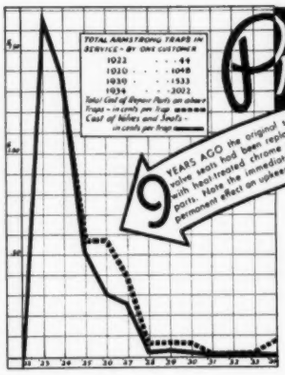
TOTAL ARMSTRONG TRAPS IN SERVICE—BY ONE CUSTOMER
1923 44
1930 1048
1930 1533
1934 3002
Proof
9 YEARS AGO the original solid valve seats had been replaced with heat treated chrome steel parts. Note the immediate and permanent effect on upkeep costs.

OF LOW UPKEEP COST

This chart is based on more than 16,500 trap years of service for a single user. The cost per trap per year (1928–1934 inclusive) averages only 37 cents for valves and seats and 48 cents for the total repair parts. No wonder Armstrong users say you can install these traps and forget them.

FOR nearly 25 years Armstrong traps have been setting a high standard for economy in maintenance. The basic design and the quality of materials and workmanship minimized the need for repairs and about the only replacements were valves and seats.

While Armstrong valve parts have always been made of the best available metal, there has been a striking improvement in this direction in the past decade. Phosphor bronze valves used to be considered good—and thousands of Armstrong traps so equipped gave fine records of trouble-free service. But under some conditions bronze would wear rapidly. Special alloys were better but not good enough for Armstrong. By constantly experimenting with heat treatment of chrome steel for the past 12 years, the present standard valve parts were developed. They have no fancy name but their uniform hardness and toughness makes them highly resistant to wear and corrosion. Their close approach to the ideal for long, hard service is best shown by facts as typified by the accompanying chart.

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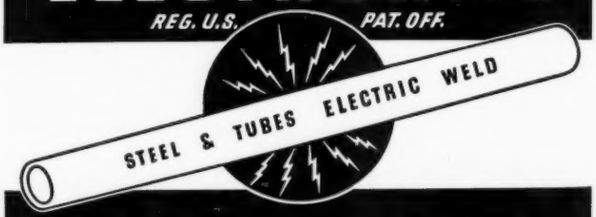

BOILER TUBES

A MODERN type boiler tube of steel or rust-resisting Toncan Iron, made from clean, flat-rolled metal formed cold to a perfect round and then welded by the electric resistance method.

The weld is as strong as the wall. Diameter, concentricity and wall thickness are absolutely uniform. Inside and outside surfaces are smooth and free from scabs, slivers and rolled-in scale. Tubes are full-normalize-annealed, soft, ductile and of uniform grain structure. Every tube is tested at pressures far in excess of code requirements.

Because of these features, Electrunite Boiler Tubes make possible tighter joints with worth while savings in time and labor, and add to the safety and life of equipment. Made in a full range of sizes for fire-tube or water-tube boilers. Write for literature.

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SUBSIDIARY OF REPUBLIC STEEL CORPORATION

STEAM ENGINEERING ABROAD

As reported in the foreign technical press

Pulverized Coal Distributor

Among the numerous exhibits at the recent British Shipping, Engineering and Machinery Exhibition described in detail in *Engineering and Boiler House Review* was a new pulverized coal distributor which formed part of the exhibit of the Department of Scientific and Industrial Research. With a view to effecting uniform distribution where the main coal and air stream must be divided into two or more branches it is first led through a number of small square ducts distributed over the whole area. Alternate rows of these ducts are inclined in opposite directions and lead into long narrow ducts which, in turn, are alternately inclined in opposite directions to take the divided streams into the final branches.

In the model exhibited two colors of balls of different sizes represented the different sized coal particles and it was observed that the quantity from each branch pipe was the same, each quantity consisting of the same number of large and small balls.

British Electrical Output 12.1 Per Cent Ahead of 1934

According to reports from the British Electricity Commission, the output of electric utilities in England for the first eight months of 1935 was 10,907,000,000 kw hr, compared to 9,733,000,000 kw hr for the corresponding period of last year. This represents an increase of 12.1 per cent. Incidently, the increase in the United States for the corresponding period was 10.5 per cent.

Peak Load Stations

The economical handling of peak loads is discussed in the October 11 issue of *Engineering* by F. de la C. Chard who refers to the quite general practice of employing the older and less efficient stations for this purpose. He points out, however, that such stations were originally designed for base load; that their boilers are usually not adapted to rapid steaming, hence high standby charges, and that the turbines as a rule cannot be loaded quickly. For these reasons the author advocates plants especially designed for peak load service. Three types of such plants are discussed, namely, steam storage, hydro storage and the installation of turbine-generators having high overload capacity through the introduction of steam at boiler pressure to the intermediate or low-pressure cylinders.

Of these the last named method shows the greatest savings but is subject to several disadvantages as it involves the construction of new primary stations and the boiler house load factor would be relatively low. The author's conclusions are that the steam storage plant, all things considered, offers the most promising method of dealing with peak loads. In support of this contention he cites the 40,000-kw steam-accumulator plant at Charlottenburg which has resulted in a boiler load factor improvement from 37 to 53 per cent, whereas the ratio of maximum to minimum load on the base-load turbines has been reduced from 19–1 to 4–1.

Nuevo Puerto Power Station

The Brown Boveri Review for October 1935 describes in detail the Nuevo Puerto Power Station of the Cie Italo-Argentina De Electricidad at Buenos Aires. This station is laid out for an ultimate capacity of 350,000 kw, but the initial installation, which has now been in operation about

five years, consists of three 35,000-kw tandem-compound Brown Boveri turbine-generators supplied with steam at 530 lb, 800 F by eight boilers. Economizers, air heaters and evaporators are provided. An exterior view of the station is here reproduced to show how the architecture differs from that prevailing among American plants.

South Africa's Largest Steam Station

The new Klip Generating Station, the largest in South Africa is now nearing completion and was described in the June and July issues of *South African Electrical Review and Engineer*. Owned by the Electricity Supply Commission the new station will be operated by the Victoria Falls & Transvaal Power Company and will have an initial capacity of 132,000 kw although an ultimate capacity of about 350,000 kw is planned. There are being installed eight boilers each rated at 180,000 lb of

steam per hour at 355-lb pressure and 735–760 F steam temperature. Each boiler will be served by an individual stack and fired by a chain-grate stoker 23 ft wide by 20 ft long. It will also have an air heater to supply the stoker with preheated air at 250 to 280 F.

There will be four 33,000-kw turbine-generators and a 7000-kw house set, each of the main turbines bleeding to an evaporator. Condenser circulating water will be cooled by large concrete cooling towers. The guaranteed steam consumption of each main turbine at full load including auxiliaries and steam bled to its evaporator is given as 350,000 lb per hr, or 10.6 lb per kwhr.

German Plant Installs Reciprocating Engines for High-Pressure Service

Employment of reciprocating steam engines for high-pressure service by a German industrial plant is described in the October issue of *Engineering and Boiler House Review* of London. A Sulzer continuous tube boiler, generating steam at 100 atmospheres (approximately 1470 lb per sq in.) and 750 F steam temperature, supplies

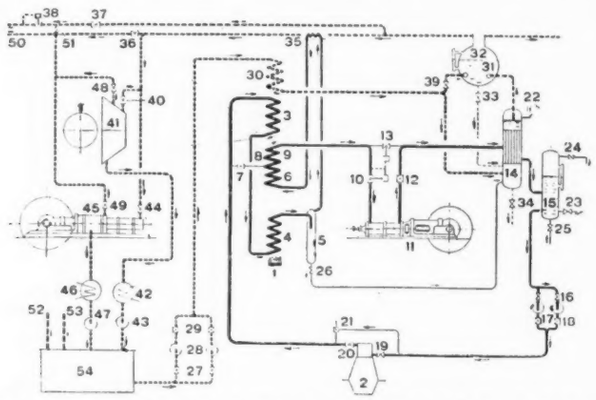


Diagram of Plant Including Sulzer Mono-Tube 100 Atm Steam Generator and Back-Pressure Engines

High-pressure Circuit.

1. Furnace.
2. Feed pump.
3. Economizer.
4. Evaporator.
5. Separator.
6. Supplementary evaporator.
7. Injection valve.
8. Injection in superheater.
9. Superheater.
10. Regulating valve for engines.
11. Back-pressure engine.
12. Non-return valve.
13. By-pass valve (automatic).
14. Steam transformer.
15. H. p. condensate collector.
16. Stop valve for pumps.
17. Feed pumps and standby.
18. Non-return valves.
19. Stop valve for recip. pump.
20. Stop valve with non-return valve for recip. pump.
21. By-pass with non-return valve.
22. Safety valve.
23. Feedwater main.
24. Oil drain valve.

25 and 26. Drain valves.

Low-pressure Circuit.

27. Stop valves for pumps.
28. Feed pumps (1 standby).
29. Non-return valves.
30. Economizer (old and new boilers).

31 and 32. Existing l.p. boilers.

33 and 34. Drain valves.

35. Superheater.

36, 37. Stop valves for circuit I and II.

38. Steam pressure reducing station for constant press. in circuit II.
39. Stop valve for l.p. boiler.

40 and 44. Adm. valves to st. eng.

41. Existing condensing steam turbine with steam extraction.

42 and 46. Condensers.

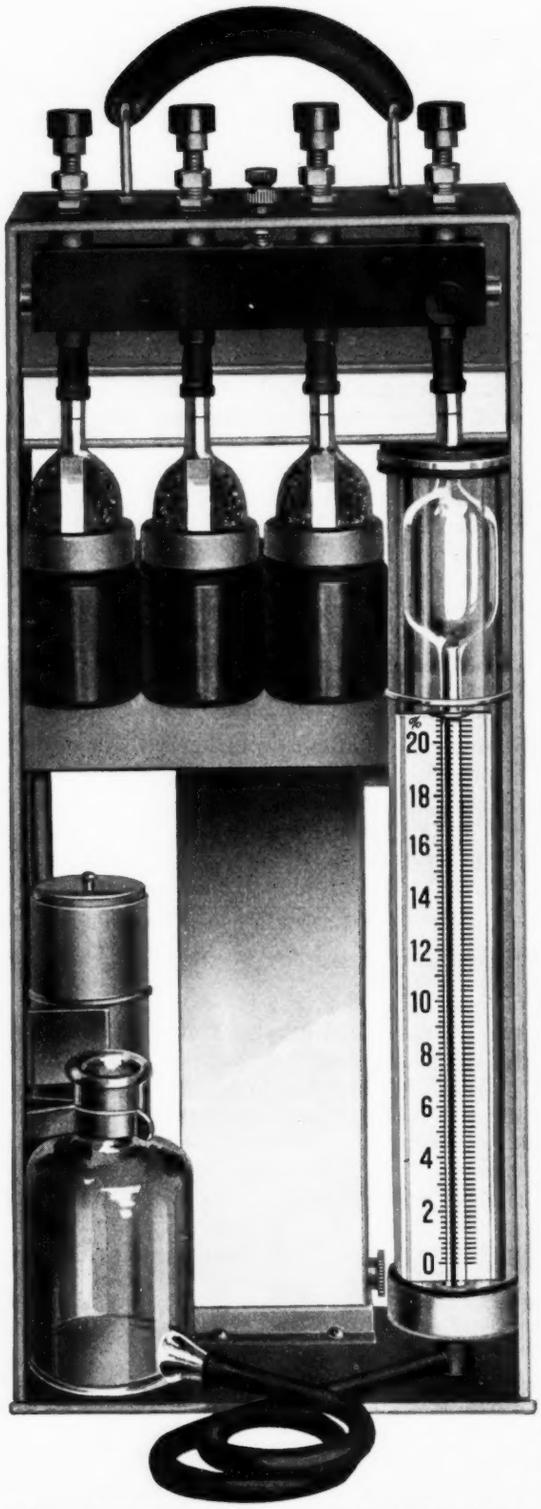
43 and 47. Vacuum pumps.

45. Existing cond. engine with intermediate extraction.

48, 49. Valves for steam extraction.

50. Circuit I with variable pressure.
51. Valve between circuits I and II.
52. Recovery of condensate, heating circuits I and II.
53. Feedwater make-up.
54. Feedwater tank.

two 800-kw steam engine-generator units which exhaust against a back pressure of about 90 lb. The exhaust from the engine passes through a heat exchanger and works on a close circuit. The hook-up which is most interesting is here reproduced. The steam engines were designed with a very flat water-rate-load curve and are said to be especially suitable for the fluctuating load under which the plant operates.



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A.S.M.E. Annual Meeting Program

This year's Annual Meeting of the American Society of Mechanical Engineers, scheduled for Dec. 2 to 6 at New York, will include an unusually large number of papers pertinent to the field of power engineering. The preliminary program just released includes the following papers of this character:

Tuesday afternoon, December 3, 2 p.m.

"Cause and Prevention of Turbine Blade Deposits," by F. C. STRAUB.

"An Airfoil Type Propeller Blower for Ventilating Electric Machine," by C. E. PECK AND M. D. ROSS.

"Influence of the Double Eddy Current on the Design of Dust Collectors," by H. VON TONGEREN.

"Cavitation Testing of Model Hydraulic Turbines and Its Bearing on Design and Operation," by L. J. DAVIS.

"Cavitation Testing on Propeller Pumps," by PROF. A. TENOT, translated by L. S. KERR.

"Design Aspect of Creep," by R. W. BAILEY.

"A Method of Analyzing Creep Data," by R. G. STURM, C. DUMONT AND F. M. HOWELL.

"The Creep of Metals," by A. NADAI AND E. A. DAVIS.

Wednesday, December 4, 9:30 a.m.

"Steam Turbine Testing" (A proposal to use Impact Pressures and Temperatures).

"The Flow Characteristics of Variable Speed Reaction Turbines," by A. EGLI.

"The Viscosity of Water and Superheated Steam," by G. A. HAWKINS, H. SOLBERG AND A. A. POTTER.

"Power Application and Other Mechanical Features in Connection with Textile Printing," by R. DEVERE HOPE.

Wednesday, December 4, 2 p.m.

"Stresses in Three Dimensional Pipe Bends," by WM. HOVGAARD.

"Creep Strength of Steels as Influenced by Microstructure," by L. L. WYMAN.

"An Experimental Investigation of Heat Absorption in Boiler Furnaces," by W. J. WOHLENBERG, H. F. MULLIKIN, W. H. ARMACOST AND C. W. GORDON.

"Critical Review of Methods of Computing Heat Absorption in Boiler Furnaces in the Light of Data Presented in Part 1," by W. J. WOHLENBERG AND H. F. MULLIKIN.

"An Empirical Method of Solving for Heat Absorption in Boiler Furnaces," by H. F. MULLIKIN.

"The Measurement of Air Flow in Fan Inlet and Discharge," by L. S. MARKS.

"The Definition of the Quantity-Head Characteristics of Fans," by M. C. STUART AND W. E. SOMERS.

Thursday morning, December 5, 9:30 a.m.

"Factors in the Selection of Coal for Unfired Stokers," by Committee of fuel engineers representing Appalachian Coals, Inc.—R. L. ROWAN, L. A. SHIPMAN, T. H. QUEER, K. J. KASPER, O. O. MALLERS AND J. E. TOBEY.

"Symposium on Zoned and Metered Air Control for Underfeed Stokers," by H. E. MACOMBER, A. S. GRISWOLD AND J. N. LANDIS.

Thursday afternoon, December 5, 2 p.m.

"The Life of Various Types of Furnace Bottoms for Tapping Ash in the Molten State," by ROLFE SHELLENBERGER.

"Water-Cooled Stokers," by J. S. BENNETT AND C. J. HERBECK.

"Electrostatic Precipitators for Stoker-Fired Boilers," by C. W. HEDBERG.

(*Continued on page 42*)

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Blow-off valves must not only be durable and efficient; but such valves must be capable of easy opening and closing in service. Strong Evertyte Blow-off Valves are especially built to meet these requirements. There is no resistance to opening or closing except pressure on the disc.

These Valves are lined with superheat packing which may be renewed under pressure.

Furnished in regular, chrome and semi-steel depending on your pressure and temperature requirements. For further information call in a Strong representative, and/ or write for Bulletin Number 100-C.

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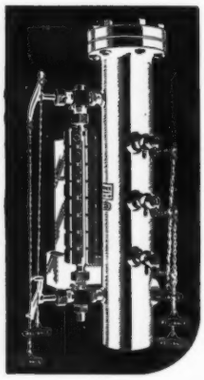

Reliance
WATER GAGE HEADQUARTERS

A.S.M.E. Annual Meeting Program

(*Continued from page 41*)

"Power Distribution Costs," by MORRIS L. COOKE.

"Film Lubrication Theory and Engine Bearing Design," by E. S. DENNISON.

"Ignition and Combustion of Diesel Fuels," by G. D. BOERLAGE AND J. J. BROEZE.

"Oil Engine Electric Generating Station Operating Costs," by GEO. C. EATON (1934 Cost Data Report.)

"The Use of Solubility Data to Control the Deposition of Sodium Sulphate or Its Complex Salts in Boiler Water," by W. C. SCHROEDER, A. A. BERK AND E. PARTRIDGE.

"Estimation of Dissolved Solids in Boiler Water by Density Readings," by J. A. HOLMES AND J. K. RUMMEL.

"Suspended Solids in Foaming and Priming of Boiler Water," by PROF. C. W. FOULK.

"Effect of Solutions on Endurance of Low Carbon Steel Under Repeated Torsion at 482 F (250 C)," by W. C. SCHROEDER AND EVERETT P. PARTRIDGE.

"Embrittlement of Boiler Steel by Caustic Soda," by G. H. WAGNER AND J. R. WALL.

"Study of Effect of Concentrated Sodium Hydroxide on Boiler Steel Under Tension," by A. S. PERRY.

"Radiation from Non-Luminous Flames," by H. C. HOTTEL AND V. C. SMITH.

"Emissivities of Refractory Materials," by R. H. HEILMAN.

"A Radiometric and Pyrometric Study of Diffusion Flames," by M. A. COOPER AND H. C. HOTTEL.

"Radiation from Pre-Mixing Burner Flames in Combustion Chambers," by E. O. MATTOCKS.

"A Study of Methods for Measuring Flame Temperature in an Internal-Combustion Engine," by A. E. HERSEY.

EQUIPMENT SALES

Boiler, Stoker, Pulverized Fuel

As reported by equipment manufacturers of the Department of Commerce, Bureau of the Census

Boiler Sales

Orders for 86 water-tube and h.r.t. boilers were placed in September

	Number	Square Feet
September 1935	86	260,585
September 1934	63	171,554
January to September (inclusive, 1935)	737	2,358,072
Same period, 1934	683	2,007,042

NEW ORDERS, BY KIND, PLACED IN SEPTEMBER 1934–1935

Kind	September 1935 Number	September 1935 Square Feet	September 1934 Number	September 1934 Square Feet
Stationary:				
Water tube	54	221,770	33	134,867
Horizontal return tubular	32	38,815	30	36,687
	86	260,585	63	171,554

Mechanical Stoker Sales

Orders for 345 stokers, Class 4* totaling 55,260 hp were reported in September by 68 manufacturers

	Installed under Fire-tube Boilers No.	Fire-tube Boilers Horsepower	Water-tube Boilers No.	Water-tube Boilers Horsepower
September 1935	275	33,148	70	22,112
September 1934	248	28,527	39	13,460
January to September (inclusive, 1935)	1,150	150,014	444	165,156
Same period, 1934	1,089	139,518	377	146,038

* Capacity over 300 lb of coal per hr.

Pulverized Fuel Equipment Sales

Orders for September not reported

STORAGE SYSTEM

	Pulverizers: Total number	Pulverizers: No. for new boilers, furnaces and kilns	Pulverizers: No. for existing boilers	Pulverizers: Total capacity lb coal per hour for contract	Water-tube Boilers: Number	Water-tube Boilers: Total sq ft steam-generating surface	Water-tube Boilers: Total lb steam per hour equivalent
September 1935	..	..	..	....	..	....	.....
September 1934	..	..	..	....	..	....	.....
January to September (inclusive, 1935)	..	..	..	....	..	....	.....
Same period, 1934	..	..	..	....	..	....	.....

DIRECT FIRED OR UNIT SYSTEM

	Pulverizers: Total number	Pulverizers: No. for new boilers, furnaces and kilns	Pulverizers: No. for existing boilers	Pulverizers: Total capacity lb coal per hour for contract	Water-tube Boilers: Number	Water-tube Boilers: Total sq ft steam-generating surface	Water-tube Boilers: Total lb steam per hour equivalent
September 1935	..	..	..	....	..	....	.....
September 1934	..	..	..	....	..	....	.....
January to September (inclusive, 1935)	..	..	..	....	..	....	.....
Same period, 1934	..	..	..	....	..	....	.....

	Fire-tube Boilers: Number	Fire-tube Boilers: Total sq ft steam-generating surface	Fire-tube Boilers: Total lb steam per hour equivalent
September 1935	..	....	.....
September 1934	..	....	.....
January to September (inclusive, 1935)	..	....	.....
Same period, 1934	..	....	.....

BOOKS

1—Dust

By S. Cyril Blacktin

296 pages Price $6.50

Dr. Blacktin has sought to supply the need for a comprehensive monograph on dust, and has written an exceedingly useful treatise. The scope includes every aspect of dust, and his book will be valued for the bibliography alone, which lists more than 500 references. Among other subjects the author deals with dust storms, explosive dusts and diseases caused by dusts.

The consideration of dust and its effects is required in widely diversified branches of science and technology, including geology, botany, agriculture, theoretical research, smelting and refining, air conditioning and many processing industries.

Dust is recommended as an authoritative reference work for research engineers, plant managers, health officials and all who have to deal with the dust problem in manufacturing, science and disease.

2—Combustion in the Power Plant (A Coal Burner's Manual)

By T. A. Marsh

255 pages Price $2.00

The author's discussion of coals and combustion is simple and understandable. His consideration of equipment—stokers, boilers, furnaces, fans and auxiliaries—is thoroughly practical. He tells how to select a stoker for the best available coal; how to design furnaces and arches; how to analyze draft problems and design chimneys, gas flues and boiler passes; how to purchase coal and calculate steam costs. He gives to every phase of his subject a practical interpretation that makes this book of exceptional value to men actually identified with steam plant design and operation.

3—Power Operator's Guide

Compiled by E. J. Tangerman

568 pages Price $4.00

This is a compilation of short practical ideas relating to power plant operation that have appeared over a period of years in *Power* under the titles of "Ideas from Practical Men" and "Right Out of the Plant." These selected communications from readers have been arranged in suitable groups under the following chapter headings: Fuels, Firing, Furnaces and Combustion; Feed Water-Handling, Treating and Heating; Boilers, Superheaters, Soot Blowers, Air Preheaters, Blowdown; Engines; Turbines and Condensers; Electricity; Oil and Gas Engines; Pumps; Air Compressors; Gages and Meters; Mechanical Transmission; Bearings and Lubrication; Piping, Valves and Fittings; Building Services—Heating, Ventilating, Air Conditioning, Hot Water; Refrigeration; Water Power; Elevators; Tools; Miscellaneous.

4—Standards on Refractory Materials

143 pages Price $1.00

In addition to all of the twenty-one specifications, test methods and definitions standardized by the American Society for Testing Materials through the work of its Committee C-8 on Refractories, this publication includes the Manual on Interpretation of Refractory Test Data, detailed information on the standard samples of type refractory materials and reports of extensive industrial surveys showing the service conditions of refractories in outstanding consuming industries.

Accurate investigation of refractory materials is difficult unless competent methods are used to analyze and report the groups of data developed. The Manual on Refractory Data, which is given, evaluates the methods of interpretation. First issued in 1932, the current Manual is extensively revised and simplified.

5—Coal and Coal Products

Compiled by F. R. Wadleigh

63 pages Price $1.00

This book of 63 pages contains not only a list of books and articles in the English language concerning coal, but also tabulates articles dealing with coal and related subjects, and similarly the publications of societies and technical institutes and bulletins and reports of various universities and colleges. The proceedings and other publications of the American Institute of Mining and Metallurgical Engineers, American Society of Mechanical Engineers, American Society for Testing Materials, American Mining Congress and the American Chemical Society are appropriately listed, as well as the reports of various commissions on coal in the United States and Great Britain.

6—Mechanics of Materials

By S. G. George and E. W. Rettger

483 pages Price $3.75

This is a simple yet comprehensive treatment of elementary mechanics of materials, written primarily for students in engineering. The book, although presupposing a limited knowledge of calculus, should also serve as a convenient reference for the practicing engineer who may be a bit rusty in this branch of mathematics. References are also given to more advanced treatises on the subject and the arrangement of the text is such that any or all of the advanced topics may be eliminated without loss of sequence in the more elementary portions. One of the distinctive features of the book is the inclusion of material on the slope deflection method for finding the deflection of a beam and the reactions of a statically intermediate beam.

7—Draft and Capacity of Chimneys

By J. G. Mingle

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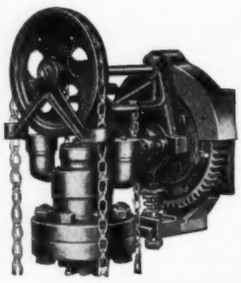

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